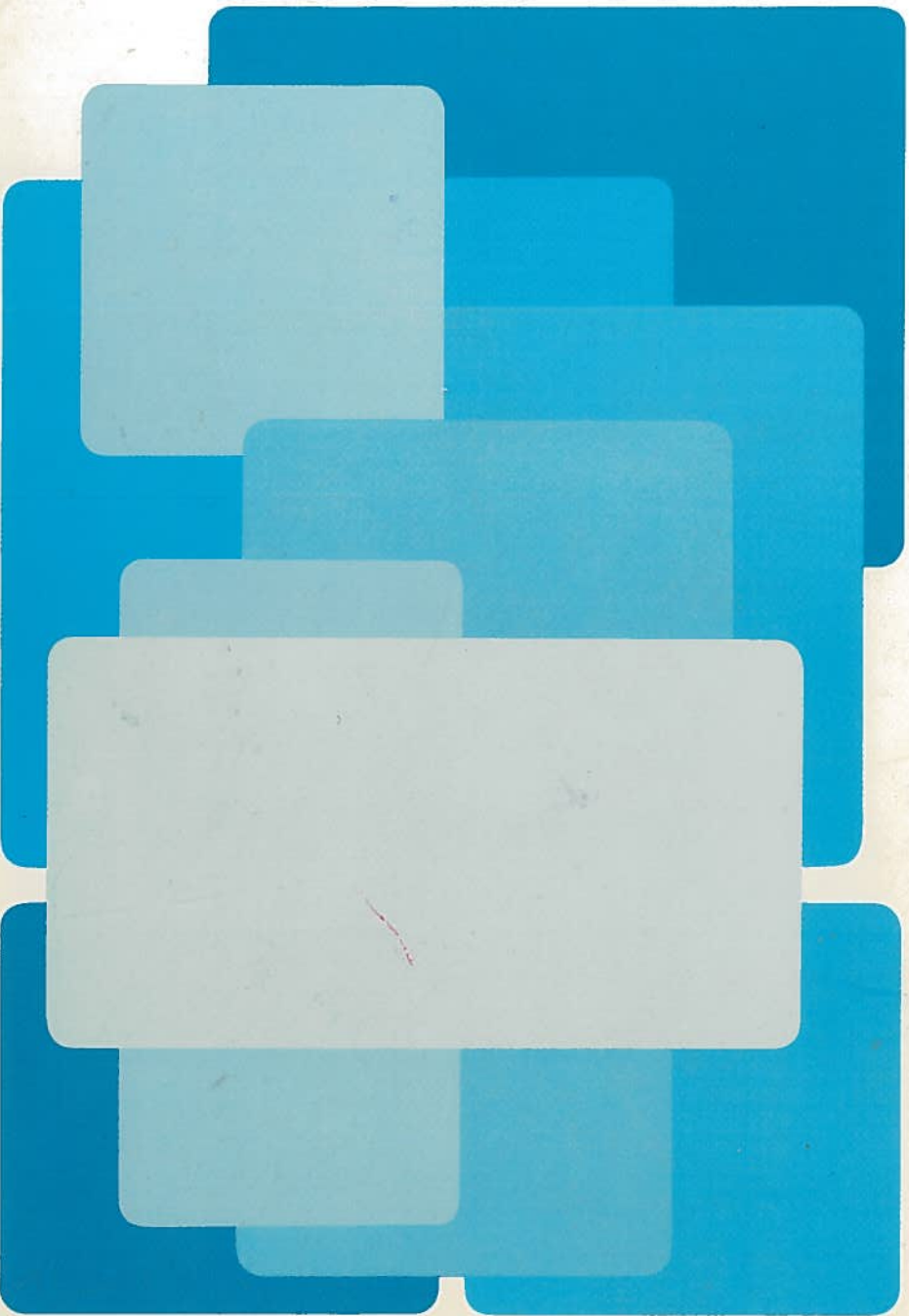


# The Hydraulic Trainer Volume 2



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Hydraulic Trainer, Volume 2

Subjects:

- Load
- Criteria
- Servo valves
- Servo valves
- Servo valves

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This textbook from Mannesmann Rexroth includes chapters covering the following subjects:

Introduction to proportional technology • Proportional valves and components • Load compensation by pressure compensators • Electronic controls for proportional valves • Criteria for the design of controls with proportional valves • Introduction into servo valve technology • Servo valves and components • From open to closed loop controls • The influence of dynamics on the closed loop control • Filtration in hydraulic installations with proportional and servo valves • Practical examples of installations using proportional and servo valves

# The Hydraulic Trainer

## Volume 2

# Proportional and Servo Valve Technology

Information and Text Book  
for hydraulic proportional and servo valves  
and also electronic components,  
as used in both open and closed loop controls

Authors

H: Dörr • R.Ewald • J.Hutter • D.Kretz • F.Liedhegener • A.Schmitt  
Mannesmann Rexroth GmbH, Lohr am Main

M.Reik

HYDAC GmbH, Sulzbach

SUNNY ENTERPRISES (PAK)  
411 Panorama Centre  
Bldg. No. 2 Raja Ghazanfar Ali Khan Road  
SADDAR KARACHI

SUNNY ENTERPRISES (PAK)  
411 Panorama Centre  
Bldg. No. 2 Raja Ghazanfar Ali Khan Road  
SADDAR KARACHI

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## Foreword

The question often asked in the seventies-

**"Is proportional technology simply a hybrid technology, bringing the power of hydraulics together with the precision and flexibility of electronic controls?" - can now, following many years of experience, be answered with a definite "Yes".**

**Proportional controls, certainly offer power and flexibility.**

Proportional valves and pumps, with their proportional solenoids, provide the ideal interface for electronic controls, and thus offer ever more flexibility in the operation of an continually growing range of machines, right through to freely programmable controls and drives.

Proportional hydraulics bridge the gap between conventional hydraulics and servo controls. They have made new possible entirely new machine concepts, both on special purpose and standard production machines.

Within a relatively short period of time, they have firmly established a clearly defined niche in the market. Their rapid advance was due, in no small part, to the fact that proportional hydraulics was initially based on conventional systems, rather than servo hydraulics. The development of electronic amplifiers of simple design, and with easily understandable functions, has simply added another dimension to the scene.

Knowledge and experience in proportional hydraulics now form the basis of successful planning of modern hydraulically driven machines. As proportional hydraulics have now invaded practically every branch of industry, this knowledge is assuming an ever increasing importance.

This book, has been designed to help those who wish to learn more about this fascinating area of technology. It is suitable for both the beginner and "climber" in proportional hydraulics, both as a text book and as a constant companion. The reader will find basic concepts clearly laid out in a similar style to that found in the "Hydraulic Trainer, Volume I".

Both proportional and servo valves are dealt, within this book, in order on the one hand to give instruction in both fields, and on the other, to show that there is no fixed dividing line between the two. At the same time, it will become clear, that proportional hydraulics are not simply "low cost servo systems". A basic knowledge of electronics is of obvious advantage in understanding the control examples shown, but in any case, the function of the electronic modules is clearly explained. In all cases, both hydraulic and electronic controls are explained in the context of examples taken from actual practice.

An entire chapter is devoted to the calculations necessary for the application of proportional controls. Again the calculation process is then illustrated in the light of a practical example.

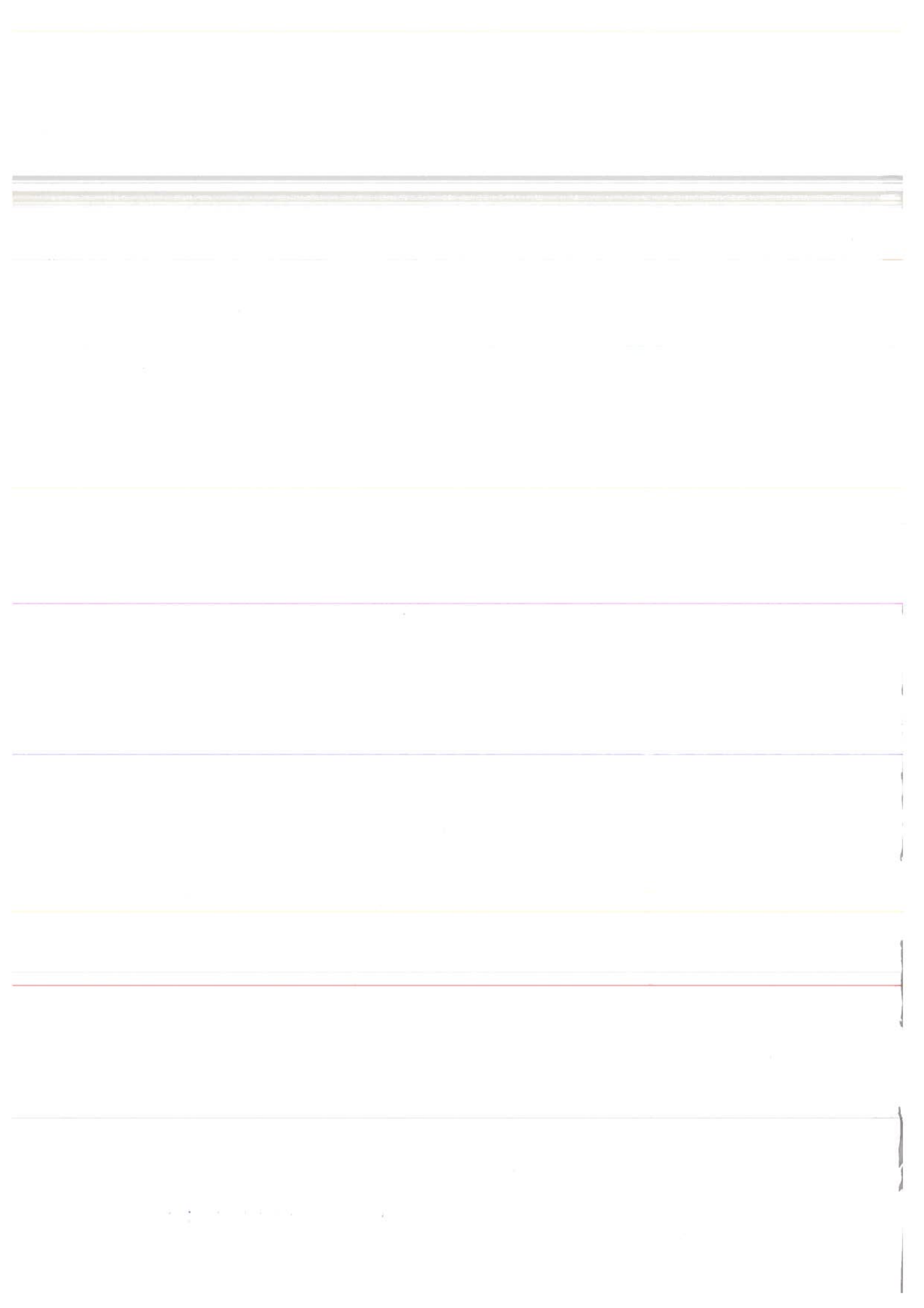


The book avoids an overbearing theoretical approach to closed loop control systems in order not to overwhelm the beginner, and cause him unnecessary stress when entering this field. The chapter "From open to closed loop control" contains a wealth of information which will enable the reader to move into practical applications. Practical examples once more round off the subject matter.

In the area of vocational training and education, the themes covering technology encompass an ever widening area. The object throughout this book has been not only introduce the subject from scratch, but also to bring the reader up to date with the latest technology.

Mannesmann Rexroth GmbH  
Lohr am Main

SUNNY ENTERPRISES (PAC)  
411 E. 10TH ST. S. Suite 100  
Bldg No 2 P. J. Ghaz of F. M. Khan Road  
SADDAR KARACHI



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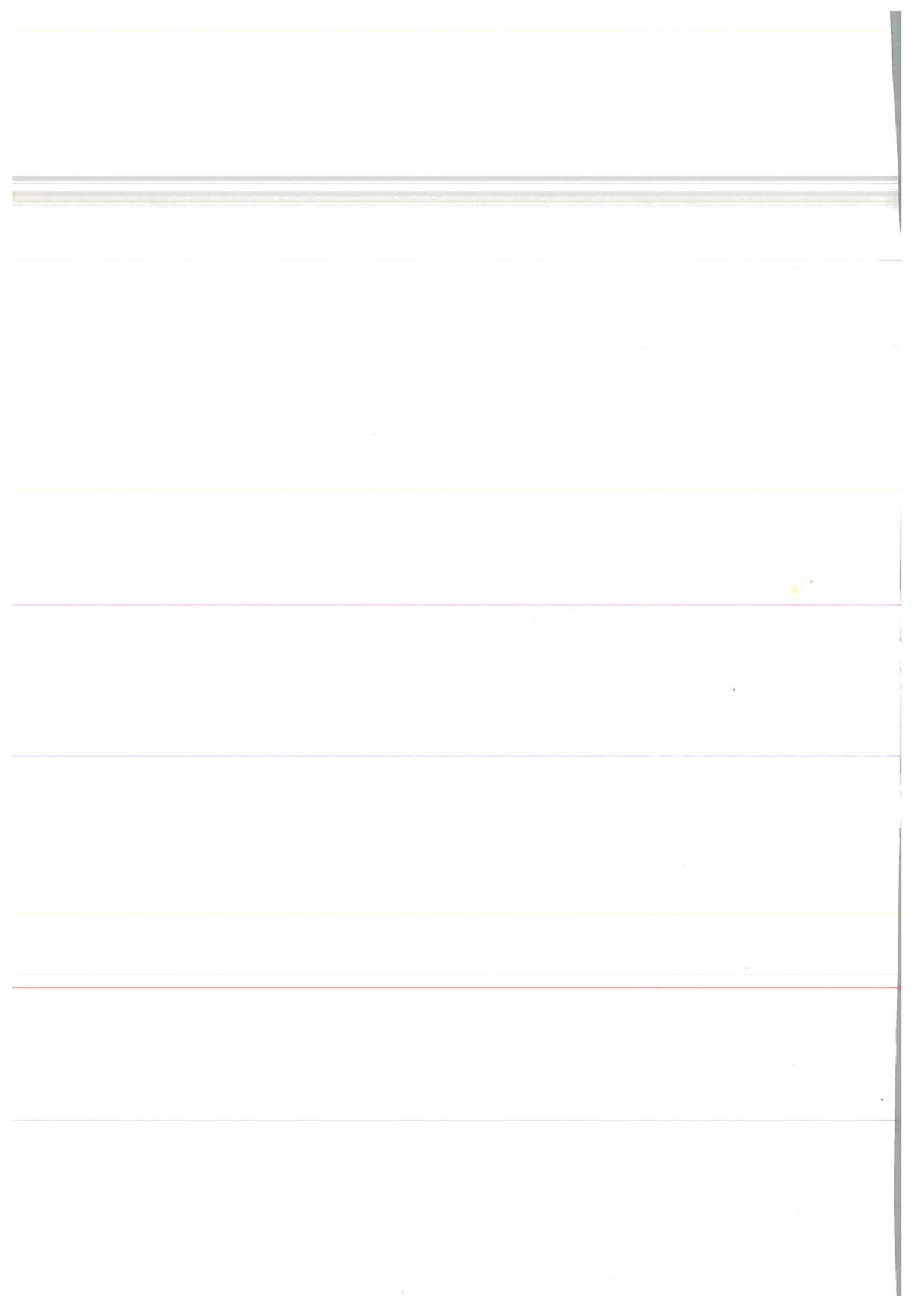
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Chapter A

# **Introduction to Proportional Valve Technology**

Arno Schmitt



## Introduction to Proportional Valve Technology

Acting as the linking element between switching and closed-loop technology, proportional valve technology has today become an established component part of hydraulic systems. Industry has been quick to implement the advantages offered by this technology.

What exactly does proportional valve technology mean in hydraulic systems?

Fig. 1 is intended to illustrate the signal sequence:

An electrical input signal in the form of a voltage (mostly between 0 and  $\pm 9$  V) is converted into an electrical current in an electronic amplifier corresponding to the voltage level, e. g.  $1 \text{ mV} = 1 \text{ mA}$ .

Proportionally to this electrical current as the input variable, the proportional solenoid produces the output variable - force and travel.

These variables, i. e. force or travel, acting as the input signal for the hydraulic valve, signify proportionally a certain volumetric flow or pressure.

For the consumer and therefore also for the working element of the machine this means, in addition to direction, steplessly variable control of speed and force.

Simultaneously, acceleration or deceleration can be steplessly varied, e. g. change in volumetric flow with respect to time.

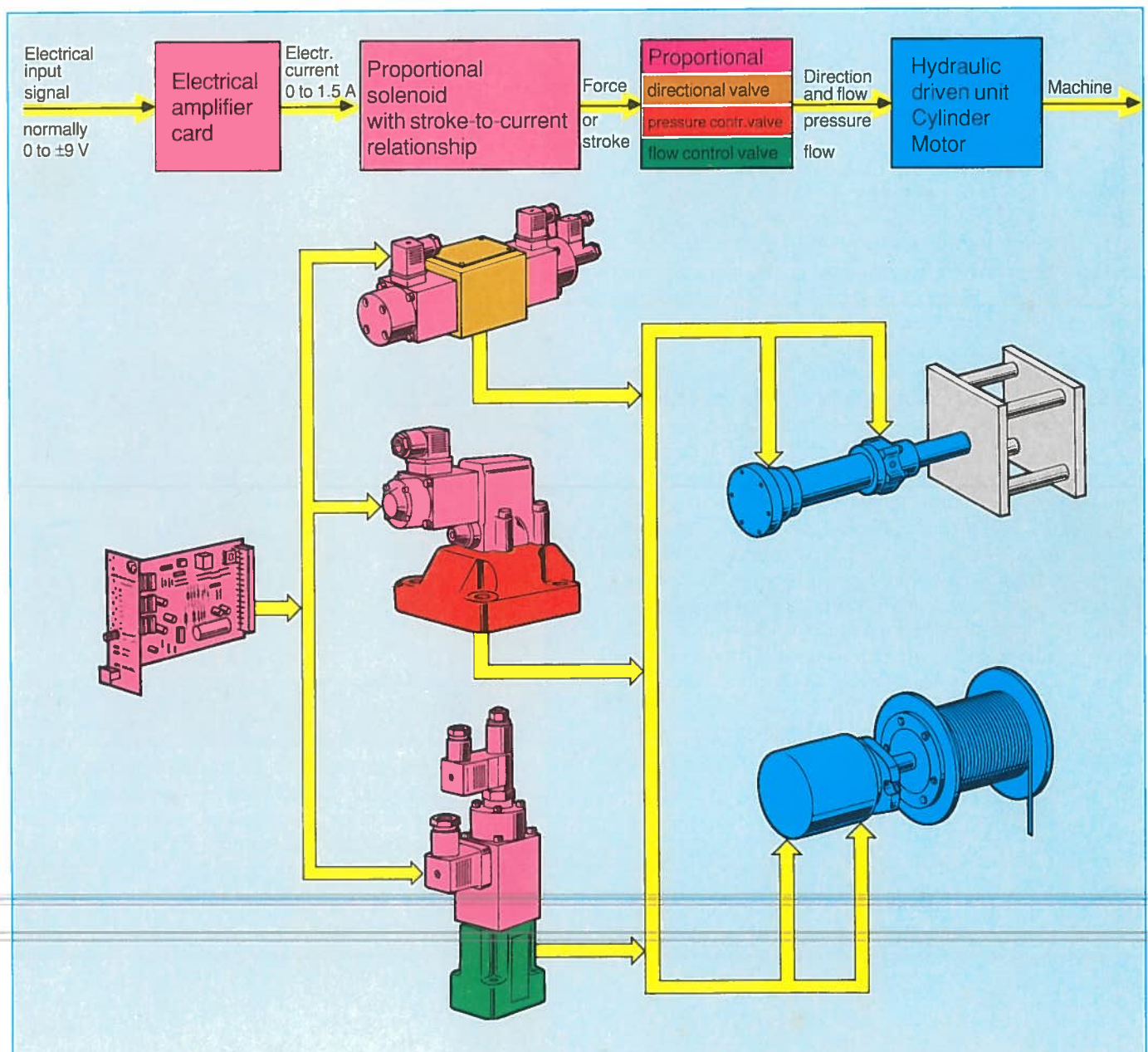


Fig. 1 Signal sequence

### An example to illustrate the possibilities offered by proportional technology.

By way of example, we will consider a welding line used in the production of motor vehicle bodies:

Individual components of the car have already been through a fast moving life during the production phase long before certain drivers can test the specified acceleration values of their cars. By analysing the diagram in Fig. 2 with regard to acceleration of body parts in a welding line, values can be obtained which, when converted, correspond to an acceleration of 0 to 100 km/h in approx. 11 seconds.

The main function of the welding line is to assemble and weld the body parts which are grouped about a platform. This production process involves several stations or better several stages.

All elevating stations are raised or lowered simultaneously in order to reach the working position, i. e. in the area of the welding tongs. The transfer of the prepared sheet metal parts to be added takes place in the middle of the lift travel range at reduced speed. The "transfer speed" must not exceed the value of 0.15 m/s, or otherwise the automatically positioned sheet metal parts would be knocked out of position. On the other hand, the lifting and lowering cycle should be completed as fast as possible, i. e. efficiently.

These requirements are met by proportional hydraulics. In the case of a solution without proportional technology, it would be necessary to, for instance, considerably reduce the maximum speed. This system would also necessitate the use of deceleration valves with corresponding mechanical cams for acceleration and deceleration, as well as flow-control valves for providing the velocity signal and, of course, directional control valves for the direction. Despite reduced acceleration and speed values, a considerably harder, less accurate and less flexible solution would be the result, whilst requiring more expensive and complex equipment.

Despite large moving masses as well as high acceleration and speed values, proportional hydraulics ensures smooth and reliable operation.



Fig. 2 One cylinder (at top) - the other one being used as a standby device - moves all stations simultaneously in conjunction with a lifting mechanism.



Fig. 3 The accumulator unit on the left-hand side provides the 460 l/min of hydraulic oil necessary for the high speed operation. The type V4 vane pump to the right fills the accumulator during the "idle periods". The proportional directional valve, type 4 WRZ 25, it can be seen lower right, on the panel.

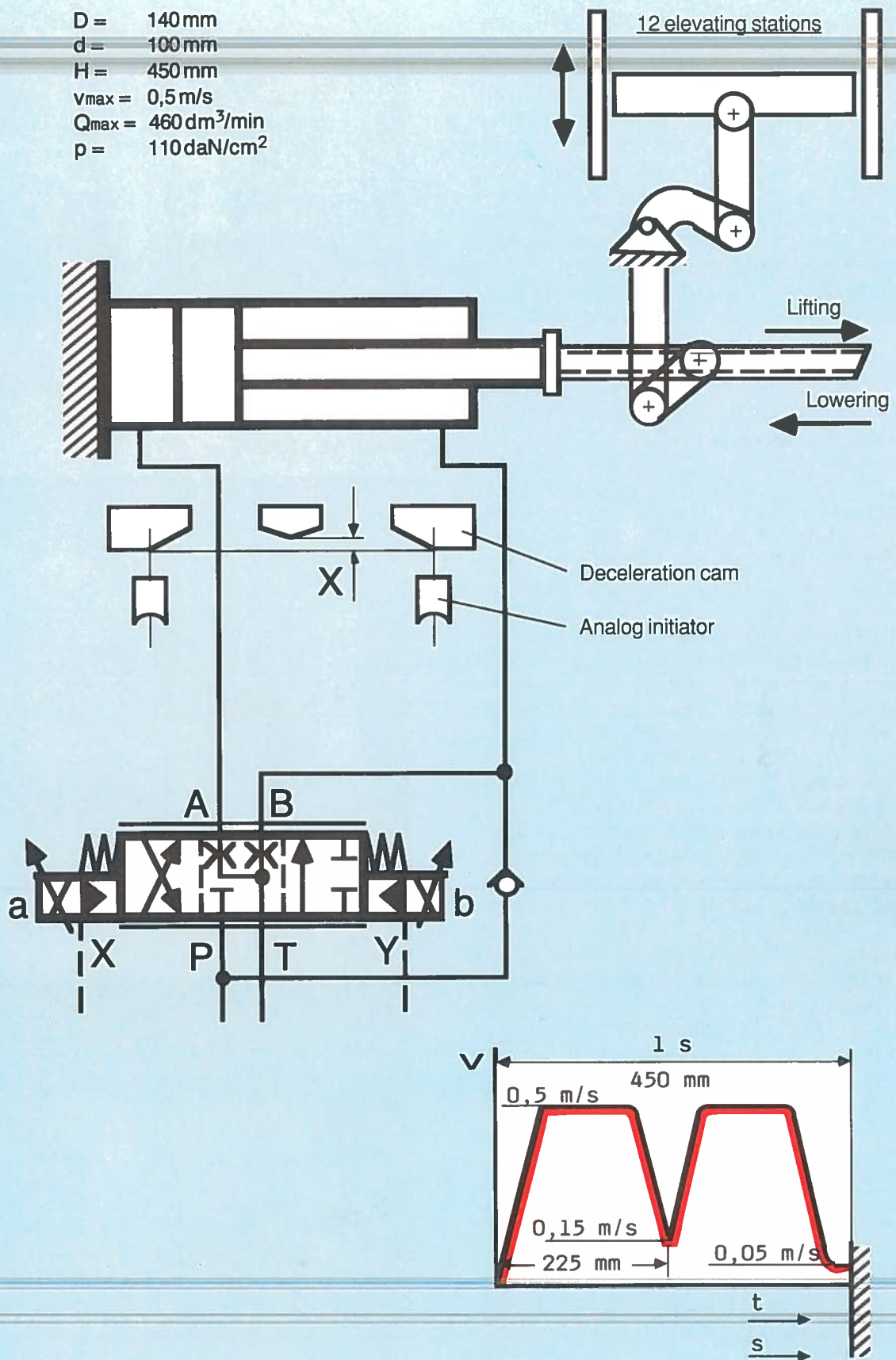


Fig. 4 Basic block diagram of a hydromechanical drive for a welding line (top) and its timing diagram (bottom right).

Proportional valves and pumps with their proportional solenoids provide perfect interface for electronic control, thereby facilitating increased flexibility in the operating cycles of production machines as well as freely programmable control systems and drives.

The technical benefits of proportional devices can primarily be found in the controlled transfer during valve change over, i. e. the infinitely variable control of signal values and the reduction of hydraulic equipment requirements for certain control applications. This therefore also represents an effective contribution to reducing material requirements in hydraulic circuits.

Proportional valves permit faster, simpler, and more precise movement cycles while at the same time improving the reversal process. As a result of controlled spool cross-over control, pressure peaks are avoided - resulting in a longer service life of the mechanical and hydraulic components.

The fact that the signals for direction and flow or hydraulic pressure are provided by electrical means has made it possible to arrange the proportional devices directly on the loads, thereby greatly improving the dynamic characteristics of the hydraulic control system.

Proportional devices in hydraulic systems found more widespread use when effective devices of simplified design were offered on the hydraulics market. These devices do not greatly differ from those of the standard hydraulic range. It has also been possible to adopt a great number of parts or assemblies from the standard range of hydraulic equipment.

The development of functionally reliable and uncomplicated standard European-format printed-circuit boards has also greatly contributed to the increased use of proportional technology.

An amplifier, containing the suitable electronic circuitry, has been designed for each type of proportional unit.

These generally include:

- voltage stabilization stage
- ramp generator
- function generator
- signal value potentiometers
- signal value relay
- pulsed output stage

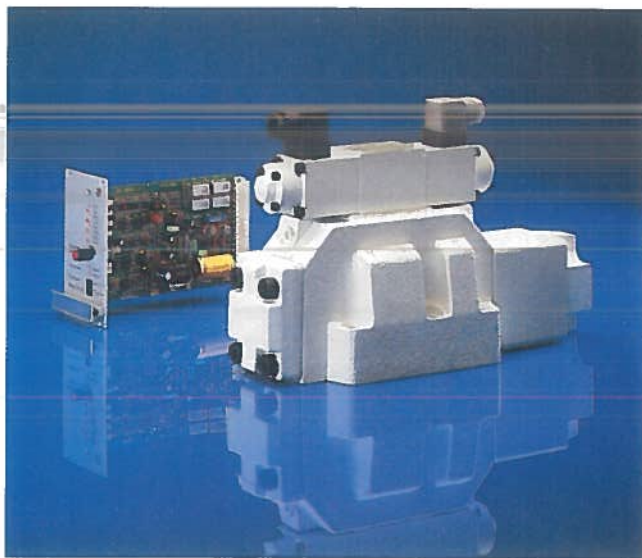


Fig. 5 Proportional directional valve, type 4 WRZ, electronic controls

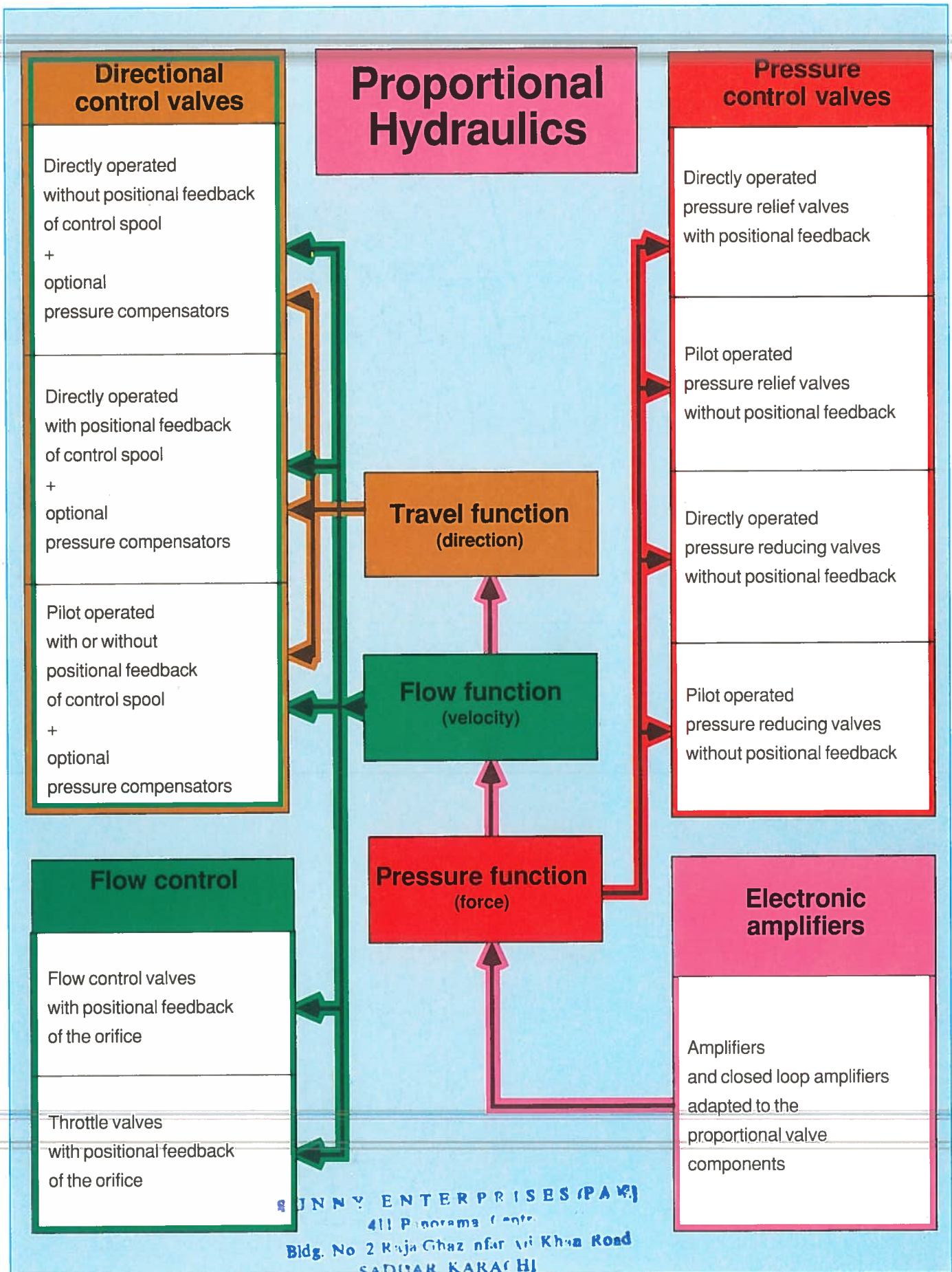


Fig. 6 Proportional pressure relief valve, type DBE, electronic controls



Fig. 7 Proportional flow control valve, type 2 FRE, electronic controls

This overview shows which functions are possible and which devices are available.



Notes

Chapter B

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## **Proportional Valves, Component Technology**

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Arno Schmitt



## Proportional Solenoids

Proportional solenoids represent the linking element between electronics and hydraulics.

The proportional solenoids are a form of DC linear solenoids. Proportional to the electrical current as the input variable, they produce force and travel as the output variable.

Corresponding to the practical application, a differentiation is made between

- solenoids with with comparatively linear force/current relationship over a reasonably long stroke length, the so-called "stroke-controlled solenoids"

and

- solenoids with particularly defined force/current relationship over a very short stroke, the so-called "force-controlled solenoids".

Only DC linear solenoids can be used for the current-proportional change in the output variables force and stroke. Due to their stroke-dependent current consumption, AC solenoids must assume their final stroke position as soon as possible.

### Force-controlled Solenoid

The solenoid force is controlled by the change in current  $I$  in the force-controlled solenoid without the armature of the solenoid performing a measurable stroke.

Due to current feedback in the electrical amplifier, the solenoid current and therefore the solenoid force are kept constant even if resistance changes.

The main feature of the force-controlled proportional solenoids is the characteristic force-stroke curve.

The solenoid force remains constant over a defined stroke range at constant current.

The stroke for the solenoid shown in this example is approx. 1.5 mm. This range is in which the solenoid is used.

The force-controlled solenoid is of compact design due to the short stroke. In view of this short stroke, the force-controlled solenoid is used particularly for pilot operated proportional directional and pressure control valves with the solenoid force being converted into hydraulic force.

The proportional solenoid is a controllable "wet pin" DC linear solenoid.

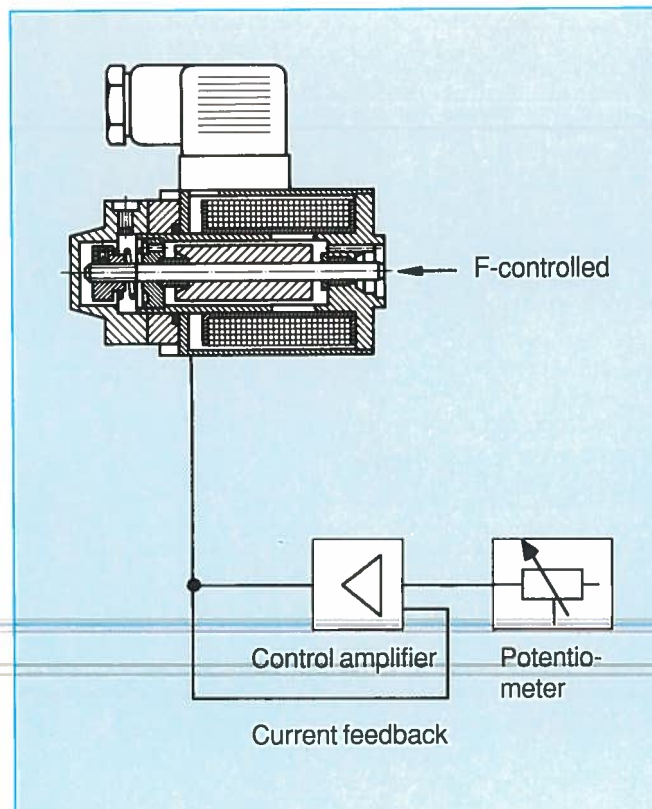


Fig. 1 Force-controlled proportional solenoid

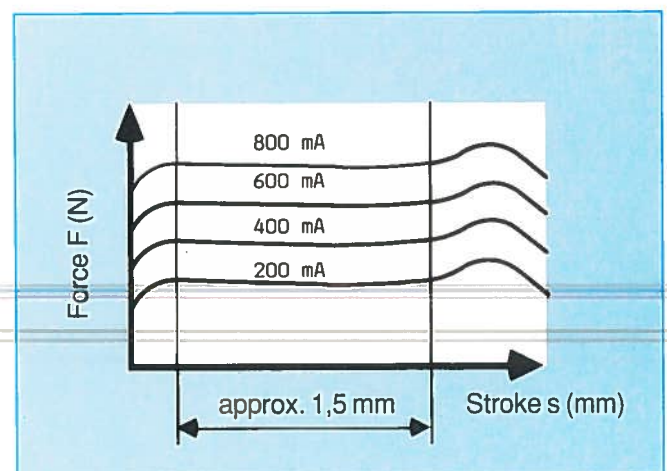


Fig. 2 Characteristic force-stroke curve

## Stroke-Controlled Solenoid

In the case of the stroke-controlled solenoid (Fig. 4), the position of the armature is controlled by a closed-loop control circuit and maintained irrespectively of the counterpressure, provided it is within the rated working range of the solenoid.

With the stroke-controlled solenoid, the spools of proportional directional, flow as well as pressure control valves can be directly operated, and be controlled in any stroke position. The stroke of the solenoid is between 3 and 5 mm depending on the size.

As already mentioned, the stroke-controlled solenoid is primarily used for directly operated 4-way proportional valves.

In conjunction with the electrical feedback, the hysteresis and the repetition error of the solenoid are maintained within very tight tolerances. In addition, any flow forces which occur at the valve spool are compensated for (relatively small solenoid force in relation to the interfering forces).

In the case of pilot operated valves, the controlled hydraulic pressure is applied to a large control area. The available positioning forces are therefore considerably greater and the percentage effect of interfering forces is not so marked. For this reason, pilot operated proportional valves can be implemented without electrical feed-back.

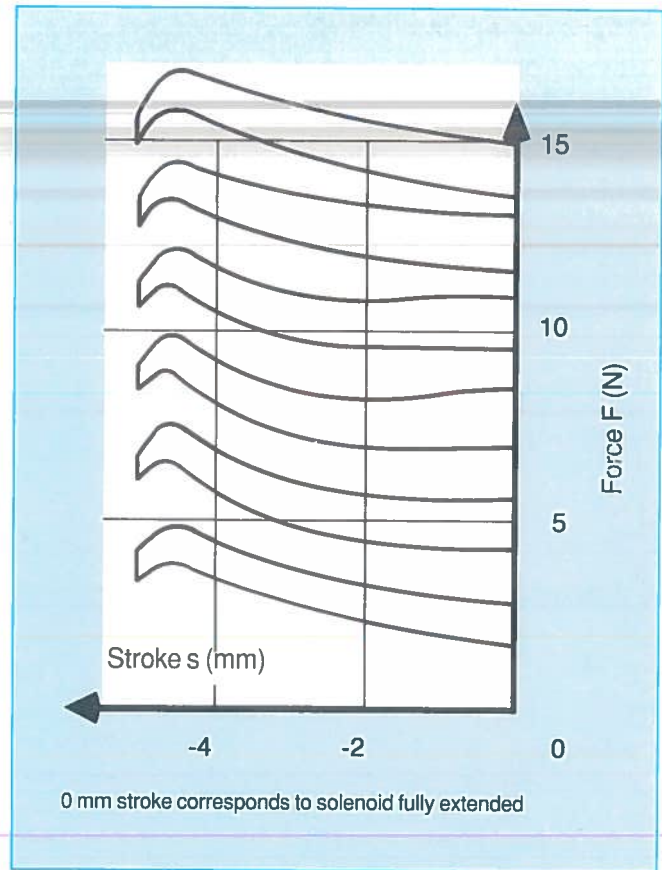


Fig. 3 Characteristic curve, stroke-controlled solenoid

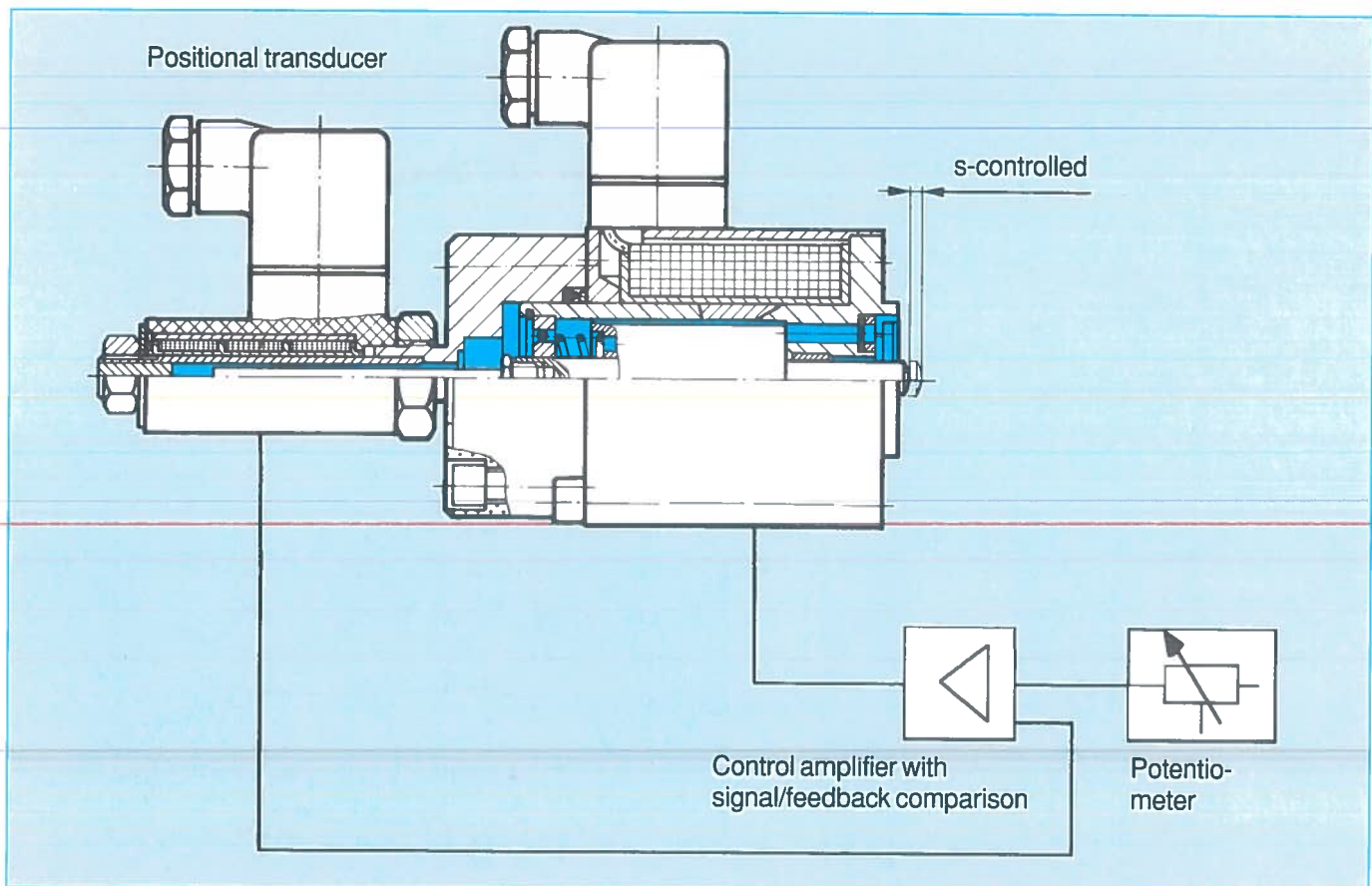


Fig. 4 Stroke-controlled proportional solenoid

## Proportional Directional Valves

A proportional directional valve is used to control the direction and value of a volumetric flow.

### Dir. Operated Proportional Directional Valve

In connection with this valve, the points applicable to the following proportional directional valves will also be discussed by way of example such as hysteresis, repetition accuracy, control spool, basic principles for characteristic curves and the time characteristics of the control spool.

The proportional solenoid acts directly on the control spool in the same way as in a conventional directional control valve.

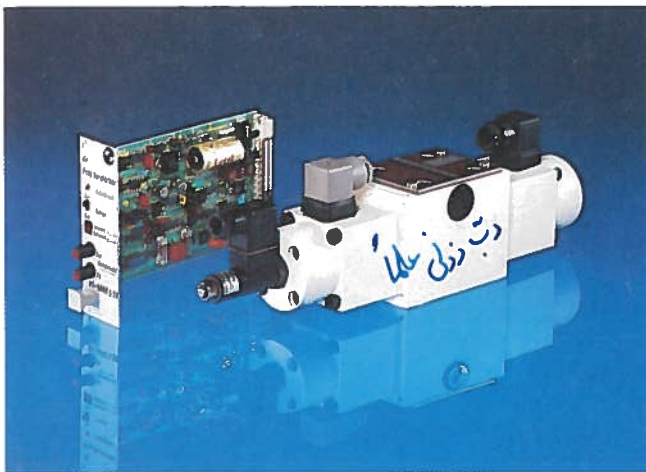


Fig. 5 Directly operated proportional directional valve Type 4 WRE 10 with electrical feedback, electronic controls

### Function

The basic components of the valve are the housing (1), one or two proportional solenoids (2) "stroke controlled" as shown in Fig. 6 with inductive positional transducer (3), the control spool (4), as well as one or two return springs (5).

The control spool (4) is held in the centre position by the return springs (5) when the solenoids are not energized. The control spool is operated directly via the proportional solenoid.

In the case of the spool shown in the figure, the link between ports P, A, B and T is closed in this arrangement. The control spool is now shifted to the right when the solenoid A for instance (left) is energized, producing the connection between P → B and A → T.

The higher the level of the signal coming from the electrical actuation system (refer to "Electronic Controls for Proportional Valves" for detailed description), the further the control spool is shifted to the right. The stroke is therefore proportional to the electrical signal. The greater the stroke the greater the opening to flow and therefore the greater the volumetric flow. The left solenoid in Fig. 6 is equipped with an inductive positional transducer to record the actual position of the control spool and to "indicate" to the electronic amplifier the position in the form of an electrical signal (Volt) proportional to the stroke.

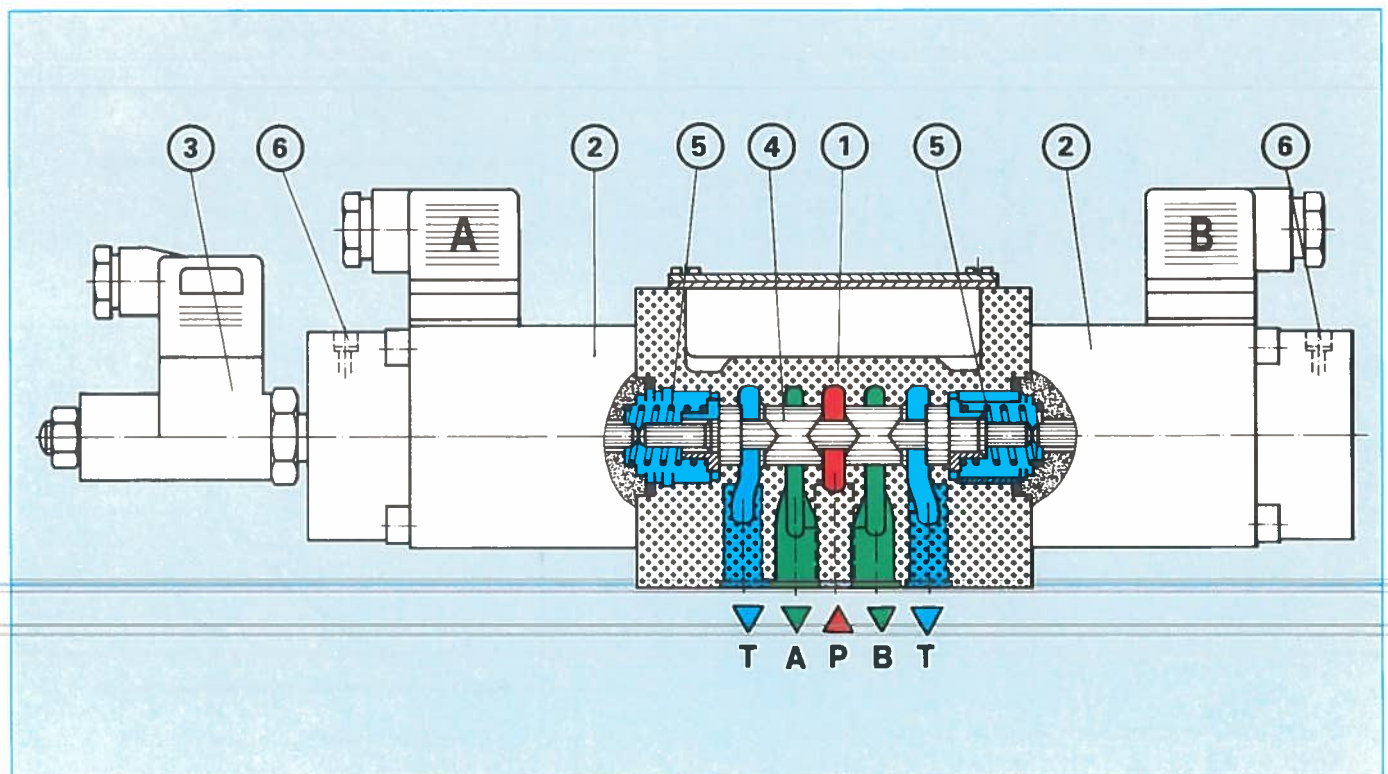


Fig. 6 Directly operated proportional directional valve with electrical feedback

Since the positional transducer is a double stroke device, spool positions both sides of centre are monitored.

In addition, the transducer is of pressure-tight design, thereby rendering a leakage oil port and additional sealing facilities unnecessary. This means that additional coefficient of friction cannot adversely effect the accuracy of the valve.

In the electrical amplifier, the actual value (actual position of the control spool) is compared with the specified value, (the signal value). This therefore represents a position control loop which detects any deviations between the specified value (signal value) and the actual value and corrects the deviations by means of corresponding signals to the relevant solenoid.

In practical applications, this means that the hysteresis and the repetition accuracy of the valve are  $\leq 1\%$  depending on the size of the valve.

**Hysteresis** general: Dependency of a condition on previous conditions.

If the electrical signal is increased from 0 to max. and decreased once again, the spool assumes a certain position proportional to the signal. The resulting deviation at constant signal value, which however, is set in various directions (coming both from the minimum and the maximum value) is termed hysteresis or hysteresis error (Fig. 8).

**Repetition-Accuracy** (repeatability)

This term describes the range within which the output signals are obtained at repeated setting of the same input signal. For the control spool, this means that a deviation of  $\leq 1\%$  with respect to the given position is achieved with repeated setting of the same signal value (for WRE).

The valve shown in Fig. 7 has no positional transducer on the solenoid. The position of the spool is therefore not additionally monitored. Once again depending on the valve size, this arrangement results in a hysteresis of 5 - 6 % and repetition accuracy of 2 - 3 %.

This degree of accuracy is completely adequate for many applications so that this version represents a relatively inexpensive solution.

**Control-Spool Design**

As can be seen in the sectional view (Fig. 6), the control spool differs from the spool in a normal directional control valve. It features triangular shaped orifice-like throttle openings, providing progressive flow characteristics (Fig. 9).

The control lands with triangular notches on the spool (Fig. 10) and the control lands of the housing constantly remain engaged with respect to each other in all positions of the spool. This means a constantly defined opening to flow in the form of a triangle.

There is therefore no position as in the case of standard directional control valves, in which these two control lands only open after an "idle stroke", and the re-engage on closing.

In addition the inlet and outlet are always throttled.

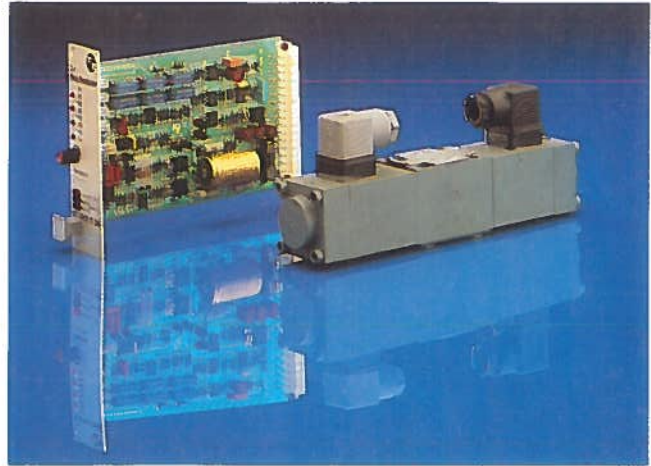


Fig. 7 Directly operated proportional directional valve Type 4 WRA 6 without feedback, electronic controls

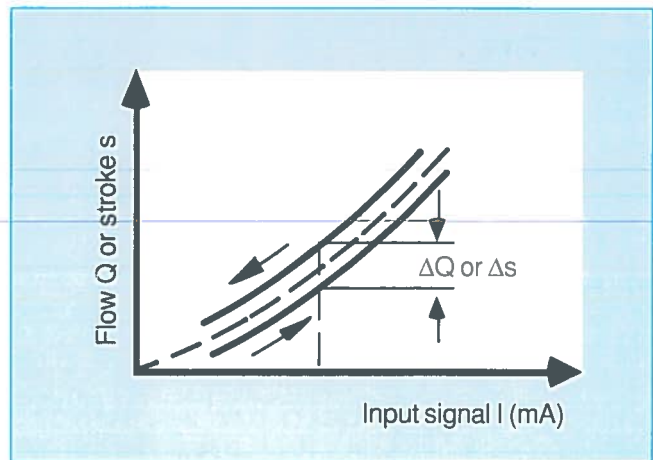


Fig. 8 Hysteresis

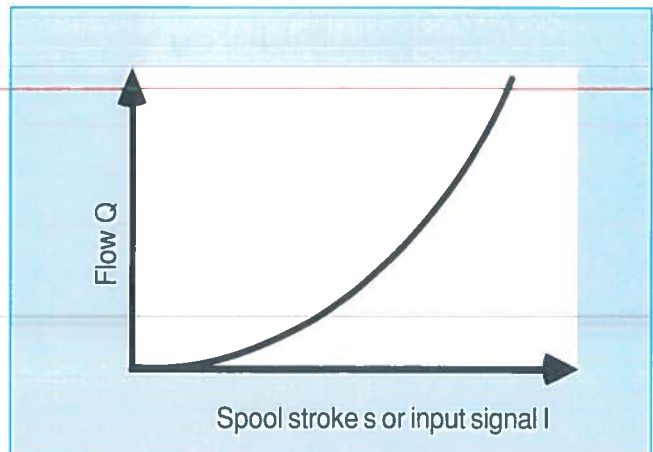


Fig. 9 Characteristic Q-h or Q-I curve

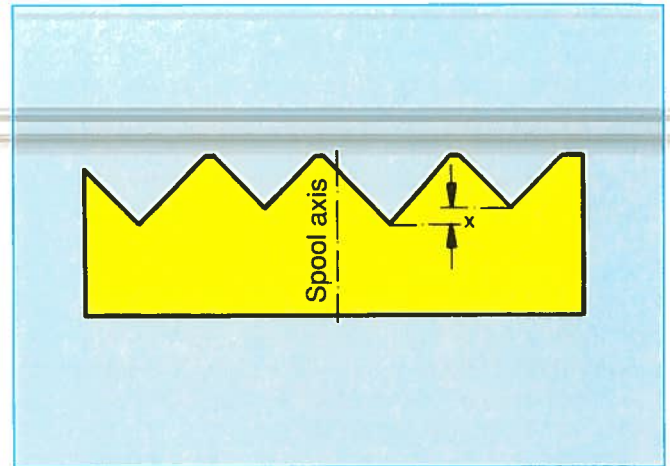


Fig. 10 Control grooves on circumference of spool having different "start" positions and long spool stroke providing excellent resolution capacity.

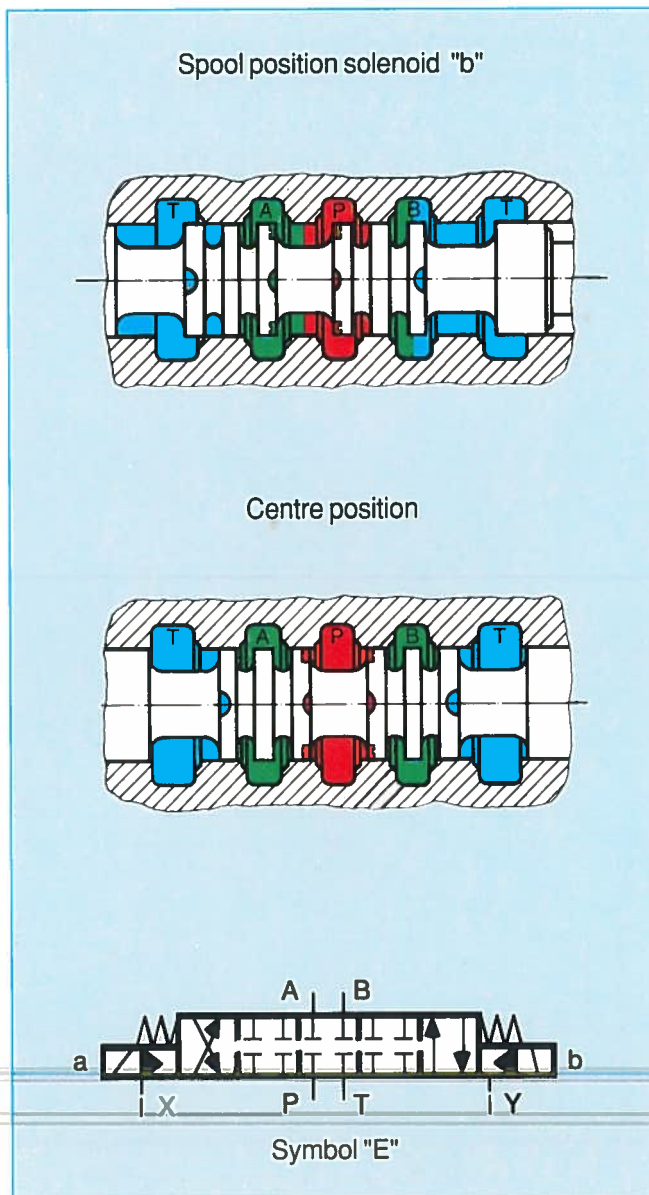


Fig. 11 Spool overlap of a standard directional valve, size 25, symbol "E" (blocked mid-position)

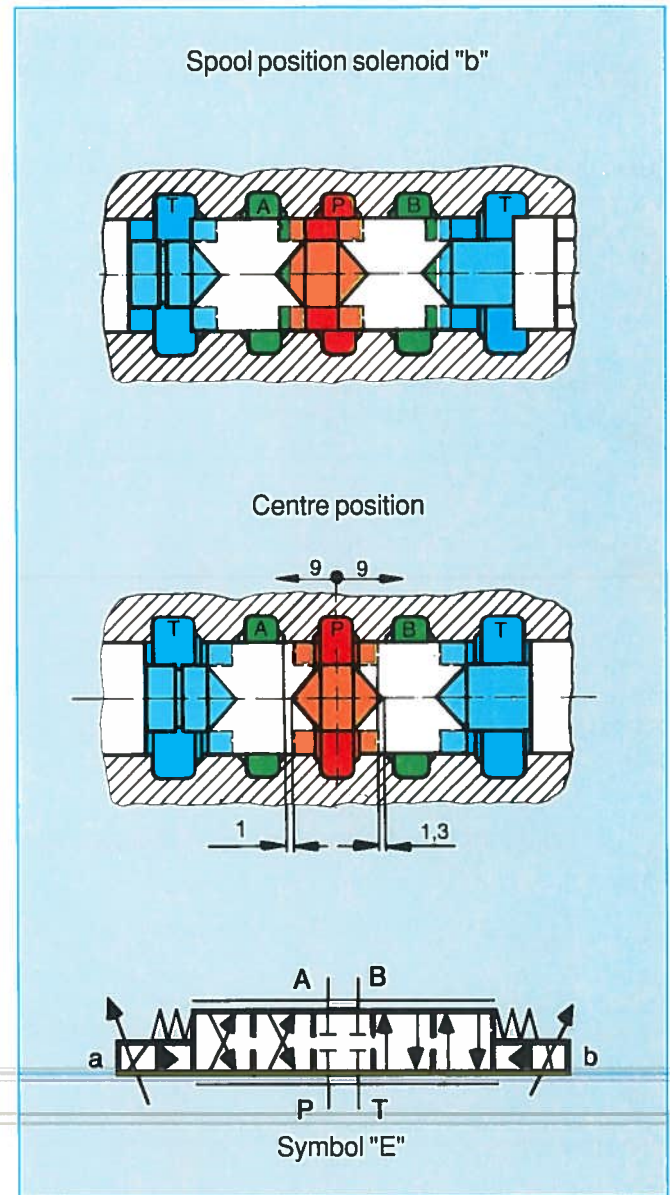


Fig. 12 Spool overlap of a proportional directional valve, size 25, symbol "E" (blocked mid-position)

## Characteristic Flow Curve

To ensure the maximum spool stroke is fully utilized, corresponding control groove openings are defined for various nominal flow rates.

The following example is intended to illustrate this statement and facilitate understanding of the characteristic curves.

The following system data are provided:

- Defined system pressure  $p = 120$  bar
- Load pressure at operating speed  $p = 110$  bar
- Load pressure at rapid traverse  $p = 60$  bar
- Required flow rate for operating speed range  $Q = 5-20$  l/min
- Required flow rate for rapid traverse speed range  $Q = 60-150$  l/min

Let us assume that a proportional valve has been selected in the same way as a standard switching valve (for  $Q = 150$  l/min nominal flow). This mistake, which unfortunately is made all too often, would lead to the following results:

- Valve pressure drop during rapid traverse  
 $p_v = 120 - 60 = 60$  bar  
 $Q_{\text{required for rapid traverse}} = 60 - 150$  l/min
- Valve pressure drop during operating cycle  
 $p_v = 120 - 110 = 10$  bar  
 $Q_{\text{required for operating cycle}} = 5-20$  l/min

## Rapid-Traversal

Referring to fig. 13, and allowing a pressure drop of 60 bar across the valve, a 66% signal allows a flow of 150 L/min, whilst 60 L/min is given by a signal of 48%. The effective control range is therefore reduced to  $66 - 48 = 18\%$  of the full range.

## Working cycle

Allowing a pressure drop of only 10 bar, then for 20 L/min, a signal of 47% is required. As a signal of 37% is required for 5 L/min, only 10% ( $47 - 37$ ) of the whole control range is available for speed adjustment. Taking a valve hysteresis of 3% (which is 30% referred to the 10% control range available), obvious difficulties in setting the speed would be encountered.

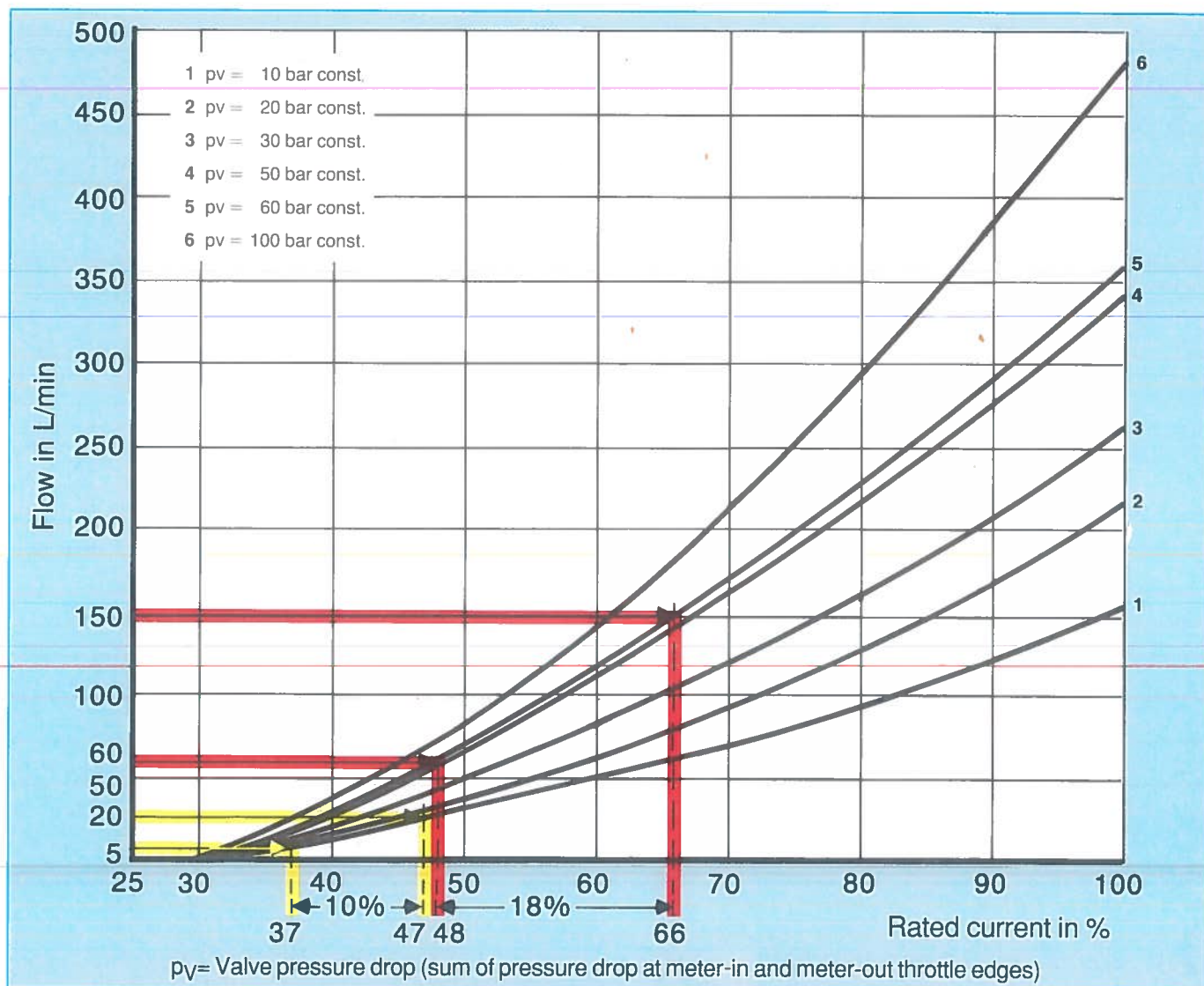


Fig. 13 Flow/rated current curve for a nominal flow rate of 150 l/min at 10 bar valve pressure drop

Referring to *fig. 14*, a correctly selected valve would, for example, be based on the following characteristic curve:

- Characteristics during rapid traverse

The signal value is between 66% and 98% (60 - 150 l/min). This provides a setting range of 32%.

- Characteristics during working stroke

The signal value is now between 36% and 63%, i.e. providing a considerably greater setting range thereby improving the resolution.

### Time Characteristic of the Control Spool

Diagrams 15 and 16 show the transfer function of the control spool with a stepped electrical input signal.

The transition from one position to another position takes place without overshoot. The spool assumes the new position in a relatively short space of time, but with a damped motion.

Thus the positioning time for acceleration and deceleration operations is more than adequate.

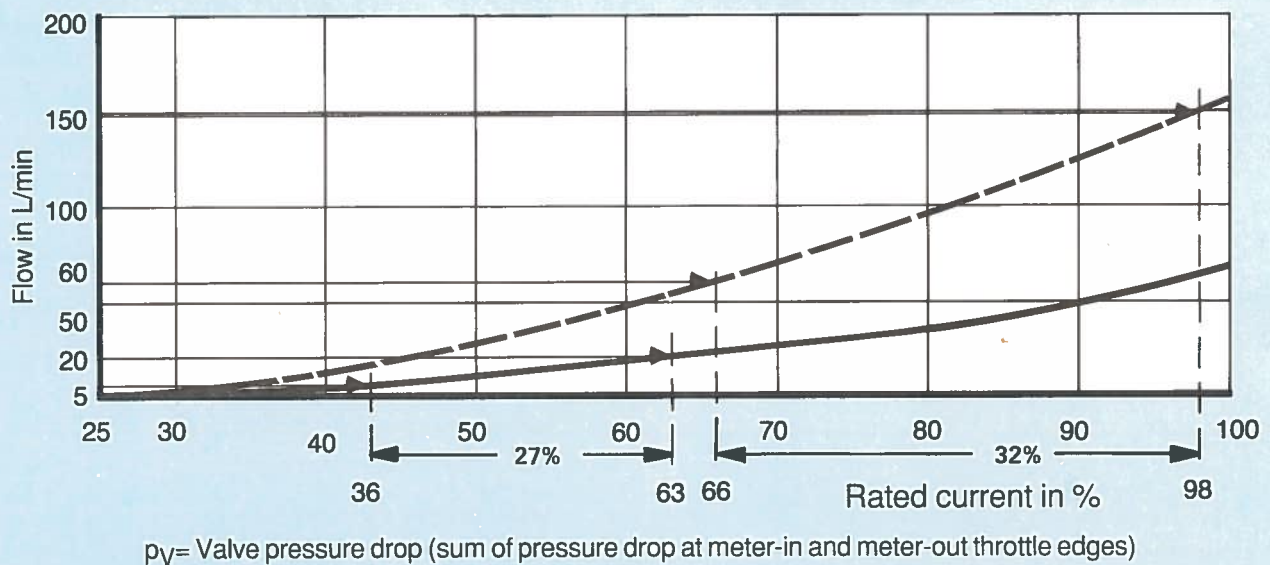


Fig. 14 Flow/rated current curve for a nominal flow rate of 64 l/min at 10 bar valve pressure drop

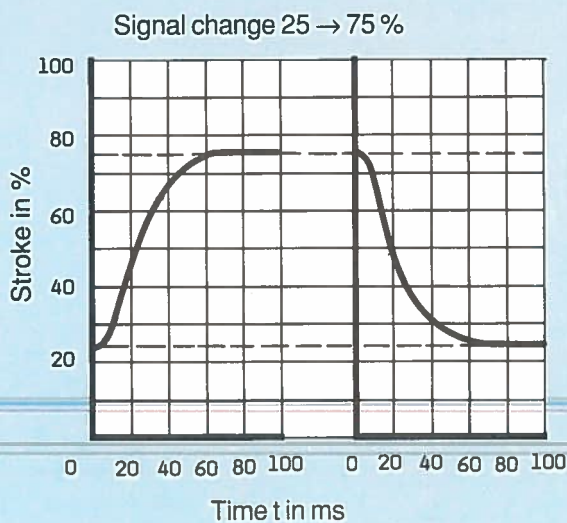


Fig. 15 Transition function with stepped electrical input signal, signal change 25-75 %

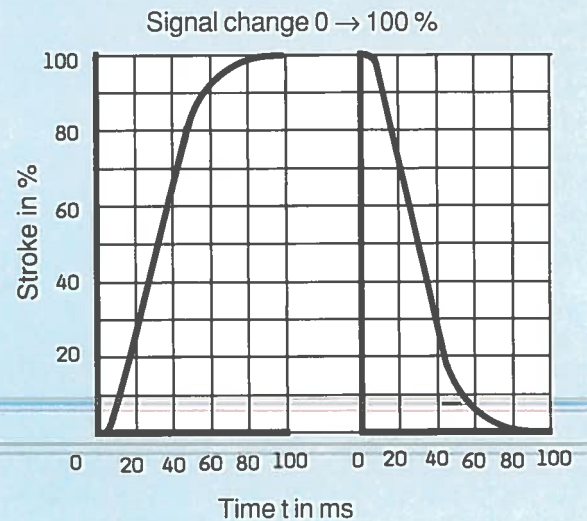


Fig. 16 Transition function with stepped electrical input signal, signal change 0 - 100 %

## Acceleration, Deceleration

In the system example described in the introduction, reference was made to the acceleration of the platform with the body parts. This acceleration, i.e. also the deceleration of a hydraulic cylinder or motor refers to the change of volumetric flow per unit of time. The positive or negative change in flow takes place via the proportional valve. The time, within which this change in flow and therefore the change in position of the control spool is to take place is preset at the electronic control for the proportional solenoid. The signal value provided by the electronics changes within the specified time to the value set as the final value.

The electrical component is termed the ramp generator, the time scale for the change in value, the ramp time.

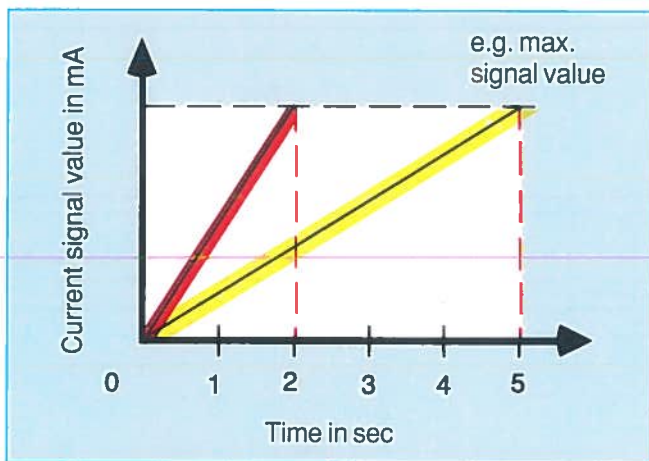


Fig. 17 Current/time diagram

- e.g. Change in current value from 0 to max. in 2 sec.  
→ short acceleration time,  
high acceleration
- e.g. Change in signal value from 0 to max. in 5 sec.  
→ long acceleration time,  
low acceleration

During deceleration, the change in signal value takes place from the high to the low value. This will be discussed in detail at a later stage in conjunction with electronic controls (refer to "Electronic Controls for Proportional Valves").

## Power-Limits

As with the standard directional control valves, the proportional valves also have power limits which should not be exceeded. The behaviour of the directly operated valve without transducer is of particular interest in this respect. Also at a large differential pressure the flow rate does not increase beyond the power limit. The spool closes itself due to the flow forces. For this reason, reference can be made here to a "natural" power limit.

The spool or valve size to be selected for a particular system in connection with flow, depends on the system pressure to be defined. This point is discussed in detail with the aid of examples in Chapter "Criteria for Control System Design with Proportional Valves".

Generally, it can be stated that a signal value of approx. 100 % should be aimed at for maximum flow.

## Control Range

The term control range refers to the ratio between minimum and maximum limits of controllable flow. For the proportional directional valve without positional transducer (Type WRA) the control range is 1 : 20. At a maximum flow rate of 40 l/min for example, the minimum flow would be 2 l/min.

Of particular significance in this respect is the repetition accuracy which, expressed as a proportion of min flow, must lie considerably below this minimum value.

For the proportional directional valve with positional feedback (Type WRE), the control range is approx. 1:100.

## Types of Spool

The following types of spool are commonly used in practical applications:

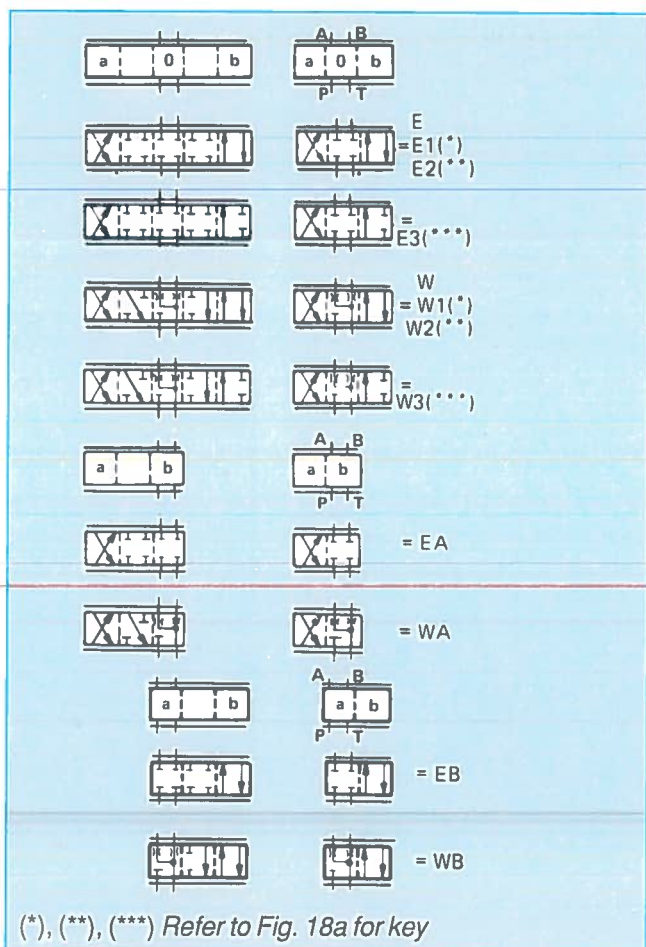


Fig. 18 Symbols with transition functions

- (\*) Symbol E1 and W1:  
 $P \rightarrow A = Q_{\max}$  |  $B \rightarrow T = Q/2$   
 $P \rightarrow B = Q/2$  |  $A \rightarrow T = Q_{\max}$
- (\*\*) Symbols E2 and W2:  
 $P \rightarrow A = Q/2$  |  $B \rightarrow T = Q_{\max}$   
 $P \rightarrow B = Q_{\max}$  |  $A \rightarrow T = Q/2$
- (\*\*\*) Symbols E3 and W3:  
 $P \rightarrow A = Q_{\max}$  |  $B \rightarrow T = \text{blocked}$   
 $P \rightarrow B = Q/2$  |  $A \rightarrow T = Q_{\max}$

Fig. 18a Flow ratios of "offset" spools

### Examples of the Individual Types of Spool

#### E-spool

The E-spool has the best deceleration characteristics. The openings to flow  $P \rightarrow A$  and  $B \rightarrow T$  as well as  $P \rightarrow B$  and  $A \rightarrow T$  are equal. It is therefore used in conjunction with double rod cylinders or, as shown in Fig. 20, with hydraulic motors.

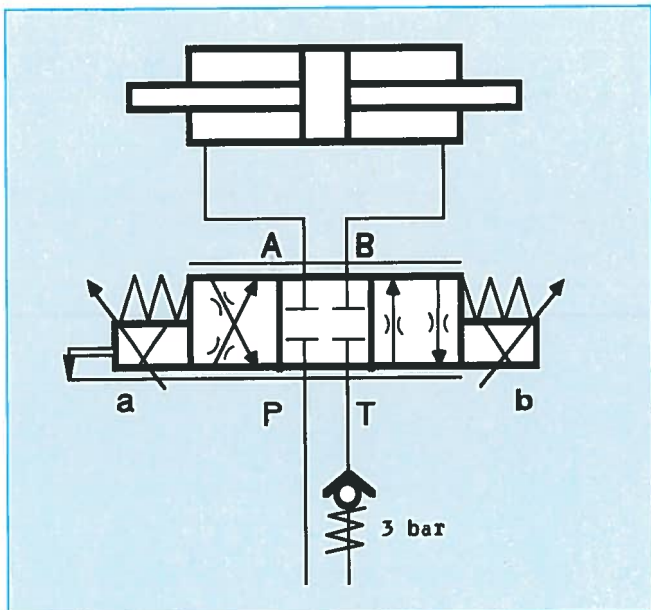


Fig. 19 E-spool with double rod cylinder

In the case of hydraulic motors, we recommend feed to the service lines as shown in Fig. 20.

Should a vacuum be created the noise level of the hydraulic motor would increase considerably.

If it is necessary to hold the motor exactly in position under load, a holding brake will be required.

If the motor is not subject to load, drifting does not occur as the result of the leakage oil at the valve, since the leakage oil of the motor is greater.

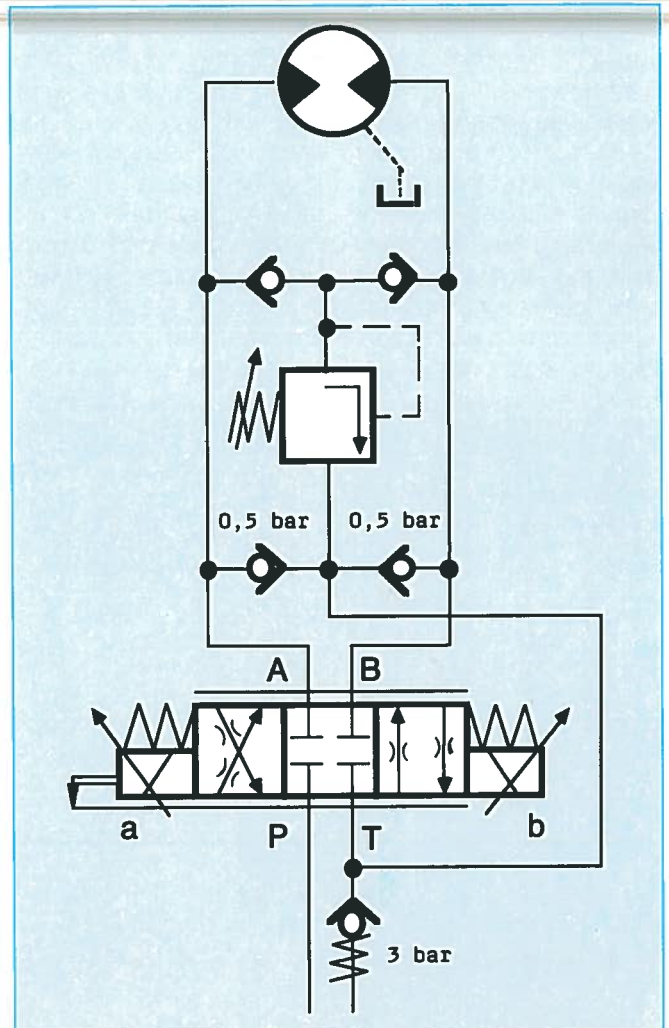


Fig. 20 E-spool with hydraulic motor

For this circuit, featuring a cylinder with a surface area ratio of  $A_K:A_R = 2:1$ , a spool should be selected with a throttle opening ratio also of 2:1. The E1-spool fulfills this requirement (also the W1-spool).

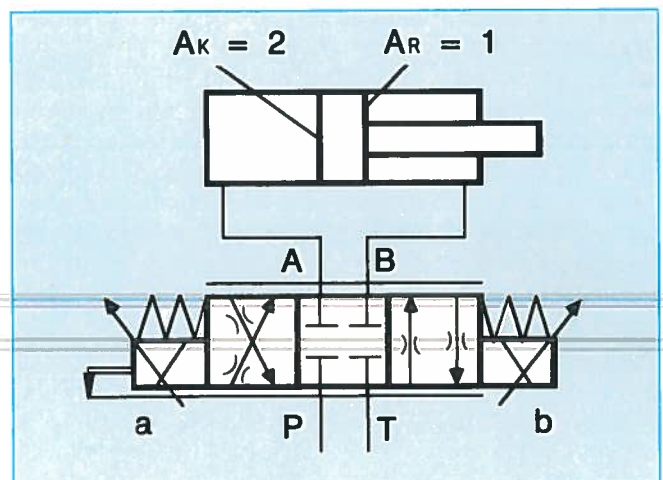


Fig. 21 E1-spool with single rod cylinder.

The following diagram (Fig. 22) illustrates the relationships. The throttle points symbolize the openings to flow in the proportional valve.

$$\begin{aligned} \text{The following applies } Q_1/Q_2 &= \sqrt{\Delta p_1}/\sqrt{\Delta p_2} \\ \text{when } Q_2 &= 2 \cdot Q_1 \end{aligned}$$

and the openings to flow are equal

$$\begin{aligned} \text{then } \Delta p_1/\Delta p_2 &= Q_1^2/Q_2^2 \\ \Delta p_2 &= (Q_2^2/Q_1^2) \cdot \Delta p_1 \\ \rightarrow \Delta p_2 &= 4 \cdot \Delta p_1 \end{aligned}$$

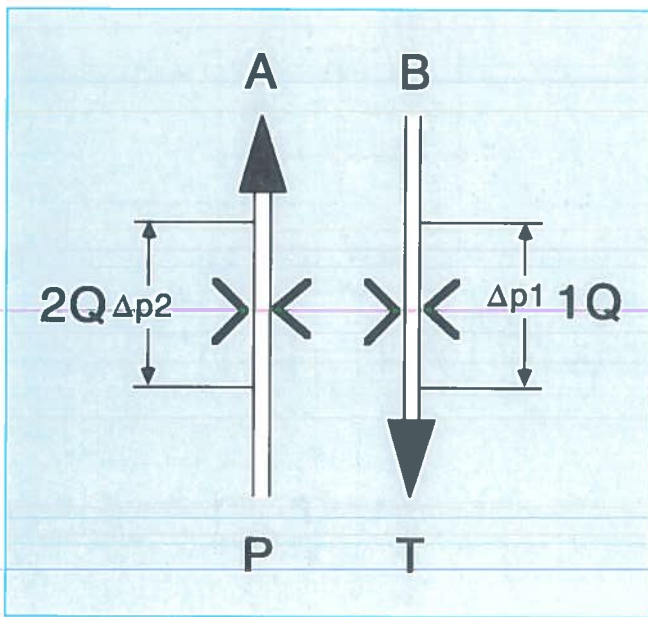


Fig. 22 Relationship of pressure drop due to flow

This relationship clearly shows that a 4-fold pressure drop is necessary in order to achieve double the volumetric flow rate at constant opening to flow.

At a ratio of the spool area to the annulus area of 2:1, and at constant throttle opening, a differential pressure ratio of 4:1 is obtained for P→A and B→T.

If the decelerating forces of gravity acting on the piston annulus require a counterpressure which exceeds the operating pressure by 25 %, then in this case it can be seen that the full bore side is not completely filled as the result of the quadratic relationship between  $\Delta p$  and  $Q$ .

These problems are avoided with the E1-spool (P→A = 1/1 opening to flow and B→T = 1/2 opening to flow) or vice versa with the E2-spool.

### E3-spool

(also refer to circuit with W3-spool)

It is used to obtain, by relatively simple means, a regenerative circuit for a cylinder with an area ratio of 2:1. The non-return valve is also possible in the form of an sandwich plate.

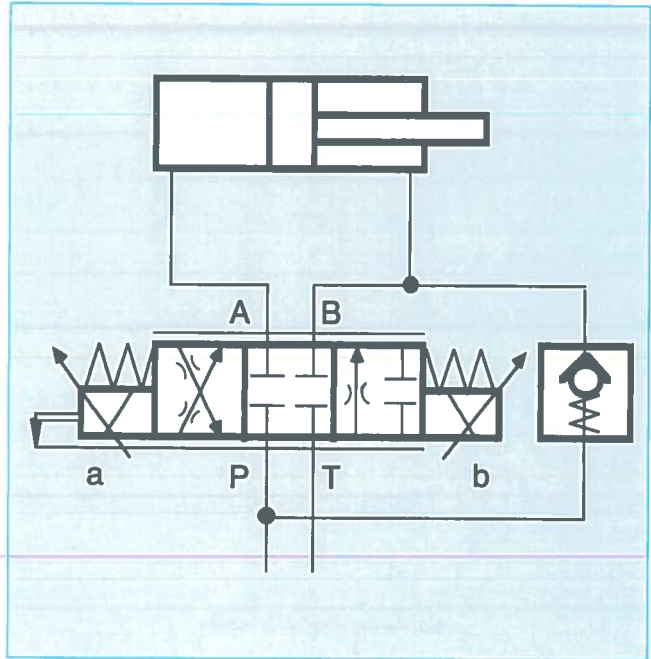


Fig. 23 E3-spool with single rod cylinder

### W-spool

In the case of a single rod cylinder and an area ratio of greater than 1:1, the W-spool prevents drifting of the unloaded cylinder as the result of leakage oil. In the mid-position, a link is obtained from A and B to T at 3 % of the nominal opening.

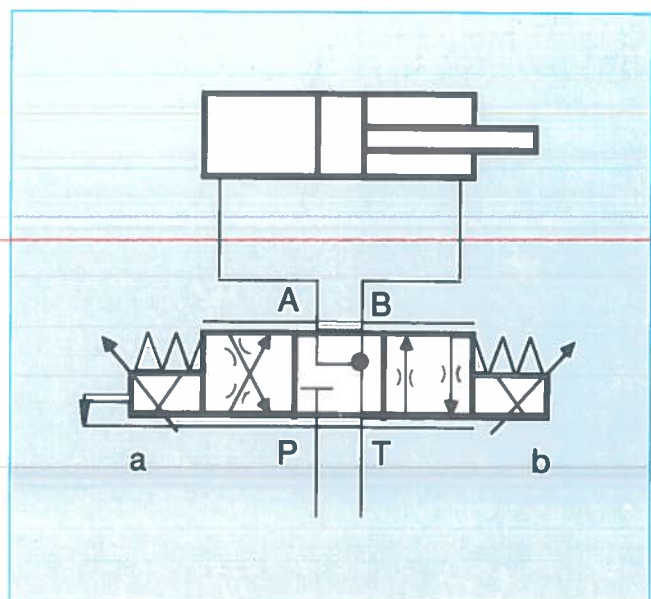


Fig. 24 W-spool with single rod cylinder

W1-spool, W2-spool

As with the E1-spool, this spool has a throttle opening ratio of 2:1 for cylinders with an area ratio of 2:1 and as the W-spool, in the mid-position it has a link from A and B to T, amounting to 3 % of the nominal opening.

W3-spool

In the same way as with the E3-spool, the W3-spool is used to achieve a regenerative circuit. In this way, the cylinder cannot spring back after deceleration since no load is applied to B - T.

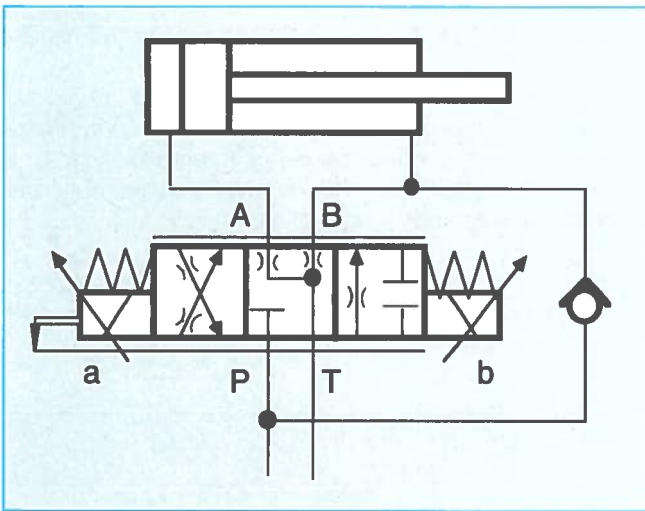


Fig. 25 W3-spool with single rod cylinder

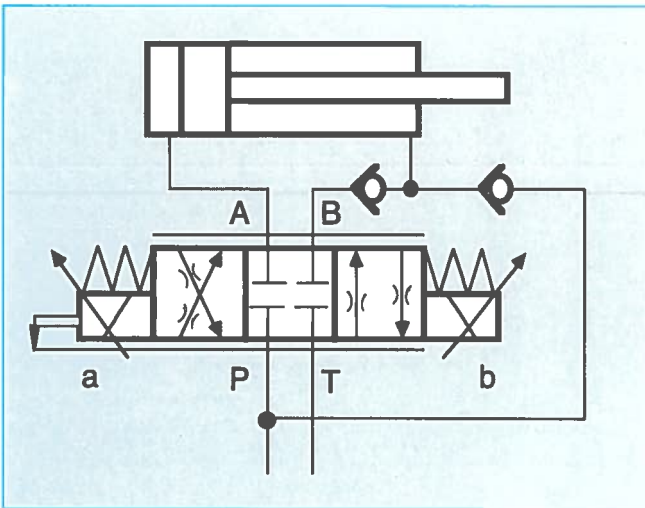


Fig. 26 Regenerative circuit with E-spool

Further Circuit Examples

Fig. 27 Single rod cylinder, area ratio almost 1:1. Vertical arrangement with weight compensation. The W1-spool is used. Weight compensation is provided by a directly operated pressure relief valve (DBDs...) with leak-free cut-off of the cylinder line.

Fig. 28 Single rod cylinder area ratio 2:1 and regenerative circuit. Vertical arrangement with weight compensation. Valve with W1-spool.

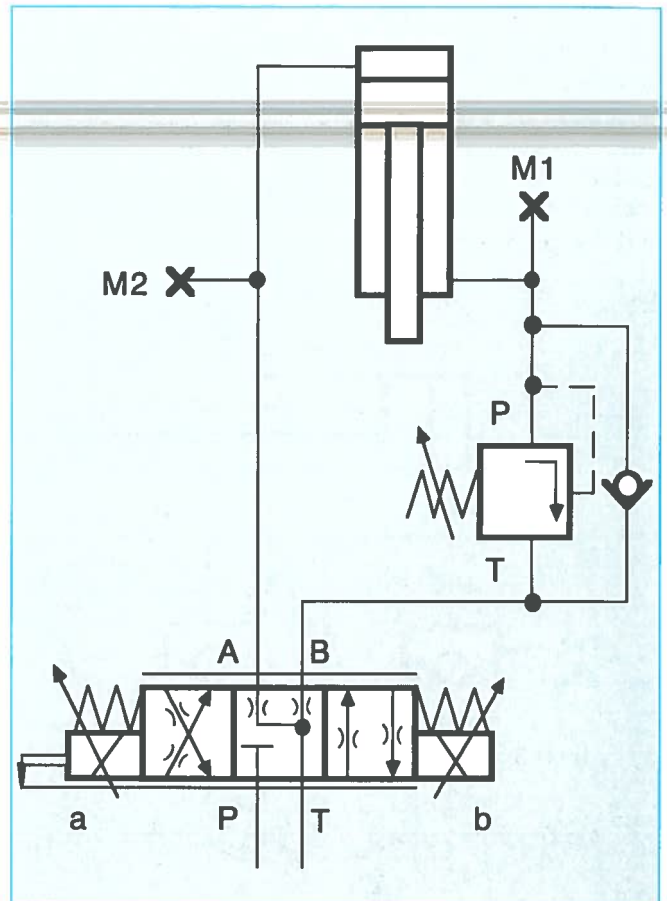


Fig. 27

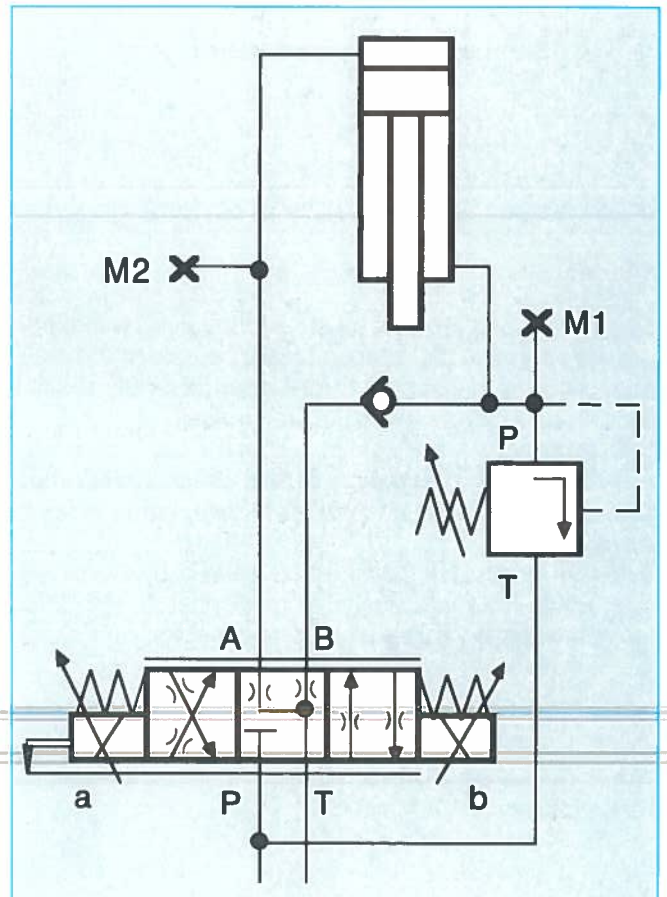


Fig. 28

### Leak-free cut-off (Fig. 29)

Due to the pressure ratios, leak-free cut-off cannot be achieved with a twinned pilot operated check valve. In this case, it is necessary to use pilot operated non-return valves with leakage oil connection. The example shows the leakage-free cut-off for both directions of movement.

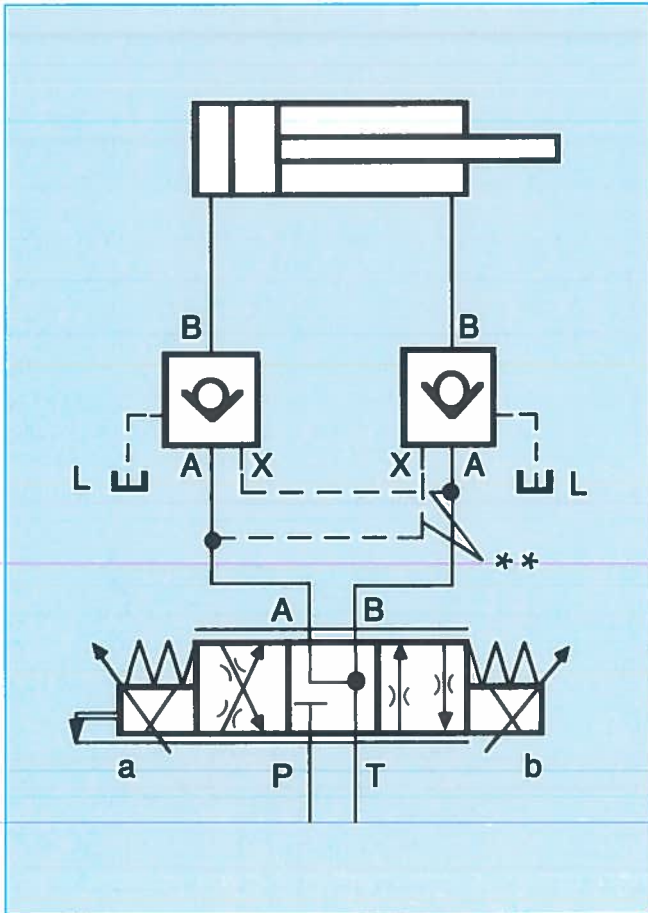


Fig. 29 \*\*

Despite the pilot operated non-return valves with external drains, particular attention must be paid to the pressure ratios in this circuit. Erratic operation may result if the pressure ratios exceed the area ratios.

\*\* Note in text

In this case, the actuation of the check valves must take place externally and not, as shown, from the opposite side.

A further possibility for cut-off is offered by the meter-out throttle isolating pressure compensator (see "Load Compensation with Pressure Compensators").

### Notes on Practical Applications

Care must be taken to ensure that the valve port A is connected to port A of the cylinder, i.e. with the piston side. This applies particularly for the E1-, W1-, E3- and W3-spools and should also be considered with regard to the basic spools since A - T represents the shortest path in the valve.

Optimum dynamic values can only be achieved when the connections between the proportional valve and driven unit (hydraulic cylinder, hydraulic motor) are kept as short as possible. Only in this way does the combination of inlet and outlet resistance, coupled by the common spool allow satisfactory control of the movement cycle.

The maximum possible acceleration of a spring/mass system, as represented by every hydraulic system, is determined by the response time of the hydraulic unit or by the spring/mass system itself.

This is illustrated with the aid of calculation examples in the Chapter "Criteria for Control System Design with Proportional Valves".

### Pilot Operated Proportional Directional Valve

As is the case with the conventional directional control valves, the larger sizes of the proportional valves are also pilot operated. The reason is the actuating forces necessary to shift the main control spool.

Normally, valves up to and including size 10 are directly operated, and pilot operated from this size upwards.

A pilot operated proportional directional valve (Fig. 33) consists of a pilot valve (3) with the proportional solenoids (1) and (2), main valve (7) with the main spool (8) and the centering spring and control spring (9).

Proportional solenoids with force/current relationship are used.

To facilitate a general view, a simplified function sequence is described in the following:

The signal coming from the electrical control is converted in the proportional solenoid (1) or (2) into a proportional force. Corresponding to this force, a pressure is obtained at the outlet (A or B) of the pilot valve (3). This pressure acts on a surface of the main spool (8) and shifts it against the spring (9) to such an extent until a state of equilibrium is obtained between the spring force and the force generated by the pressure. The stroke of the spool and therefore the opening to flow depends on the pressure head acting on the surface of the spool. The hydraulic pressure produced by the force controlled solenoid can be produced using either a pressure reducing, or a pressure relief principle.

The valve described in this example is equipped with a pressure reducing valve as the pilot valve. The advantage lies in the fact that pilot oil does not flow continuously through the valve.

The 3-way pressure control valve (Fig. 30) basically consists of two proportional solenoids (1) and (2), housing (3), a control spool (4) and 2 pressure measuring spools (5) and (6).

The proportional solenoid converts an electrical signal into a proportional force, i.e. an increase in the control current results in a correspondingly higher solenoid force. The set solenoid force remains constant over the entire control stroke.

When the solenoid is not energized, as shown in Fig. 30, the control spool (4) is held in the mid-position by the springs. The ports A and B are connected to port T and therefore not under pressure. Port P is closed. If, by way of example, solenoid B (1) is energized, the force of the solenoid acts via the pressure measuring spool (5) on the control spool (4) and shifts it to the right. As a result, oil flows from P to A. As before, port B remains linked to T. The pressure building up in port A acts on the pressure measuring spool (6) via the radial hole in the control spool (4). The resulting force generated by the pressure opposes the solenoid force and shifts the control spool (4) in the close direction when a balance is reached between both forces. During this

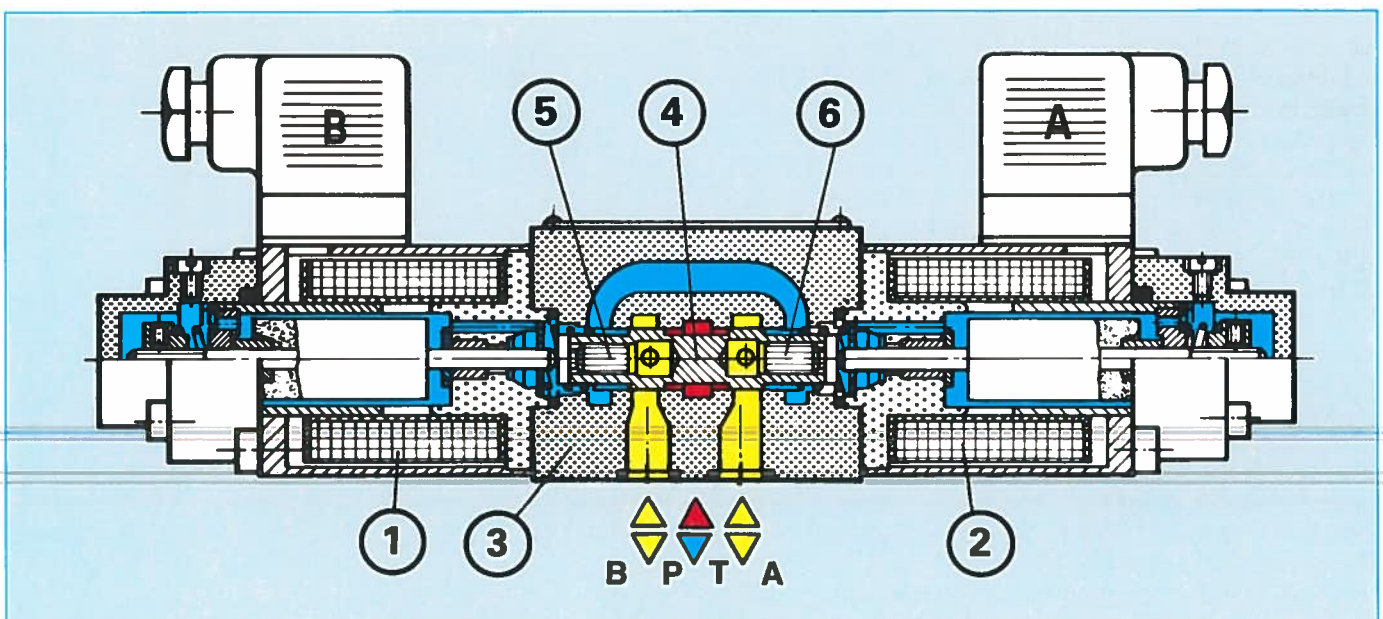
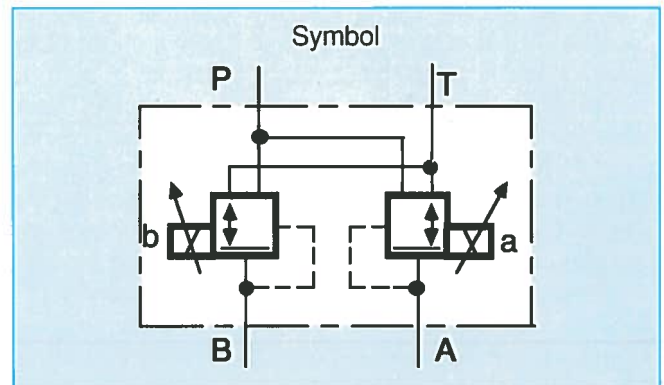


Fig. 30 3-way proportional pressure control valve Type 3 DREP 6 used as a pilot valve

procedure, the measuring spool (6) rests on the plunger of the solenoid (2).

The link from P→A is interrupted, the pressure is held constant in the working port A. A reduction in the solenoid force results in excess pressure force at the control spool (4) so that it is shifted to the left.

Pilot oil can flow via the connected ports A→T so that the pressure can drop correspondingly.

Equilibrium of forces once again means that the pressure is maintained constant, however, now at the lower level.

In the neutral position - proportional solenoid deenergized - the ports A and B are opened to T, i.e. oil can flow unrestricted to the tank and pressure is relieved at B or A. At the same time, the link P→A or P→B is interrupted.

With the aid of the pilot valve, we can therefore vary the pressure in the ports A or B proportional to the electrical input signal.

If the chambers (10) and (12) are depressurized, i.e. A and B of the pilot valve, the main spool (8) is held in the centre position by the centering spring (9).

### The Effect on the Main Spool

If, by way of example, solenoid B energizes once again, pilot oil is allowed to flow either internally from the channel P or externally through the port X via the pilot valve to the chamber (10). Here pressure is built up proportional to the input signal. The resulting force generated by the pressure shifts the main control spool (8) against the spring (9) (Fig. 33a) until the spring force and pressure force are in equilibrium. The value of the pilot pressure therefore determines the position of the spool which in turn determines the size of the orifice-type opening and therefore the flow rate.

The design of the main control spool corresponds to that of the directly operated proportional directional valves.

If solenoid A (2) is energized, a pressure builds up in the chamber (12) corresponding to the signal. This pressure once again shifts the main spool (Fig. 33b) against the spring (9) via the tie-rod (13) which is rigidly linked to the spool.

The spring (9) is preloaded between the thrust pads and fitted without play between the cover and housing.

The use of **one** spring for both spool directions ensures an identical valve reaction in each direction to any given signal, and thus, equal deflection in each direction. In addition, the thrust pad mounting system allows a particularly low-hysteresis to be achieved.

The spring once again forces the control spool into the centre position when the pressure is relieved in the pressure chamber. The facilities for pilot oil feed (internal or external) as well as for pilot oil outlet (internal or external) are the same as those for conventional pilot operated directional valves.

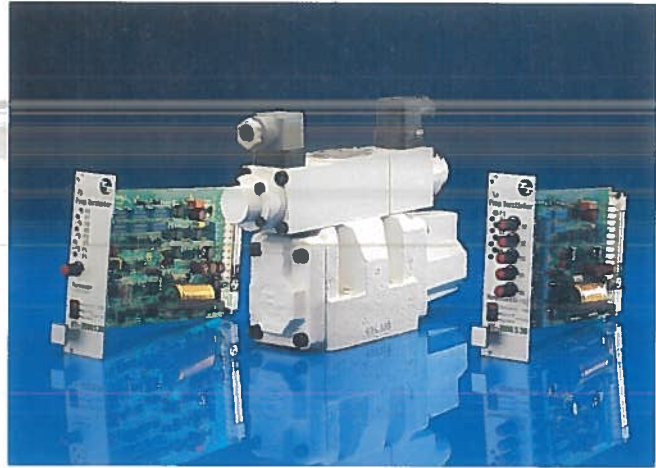


Fig. 31 Pilot operated proportional directional valve Type 4 WRZ, electronic control

The required pilot pressure is  $p_{st \min} = 30 \text{ bar}$  and  $p_{st \max} = 100 \text{ bar}$ .

The hysteresis is 6 %.

The repetition accuracy is 3 %.

The characteristic response curve with stepped electrical input signal shows also in this case that the control spool takes up its new position without overshoot (Fig. 32). This is due to the strong centering spring. As a result, flow forces also have no effect on the spool position.

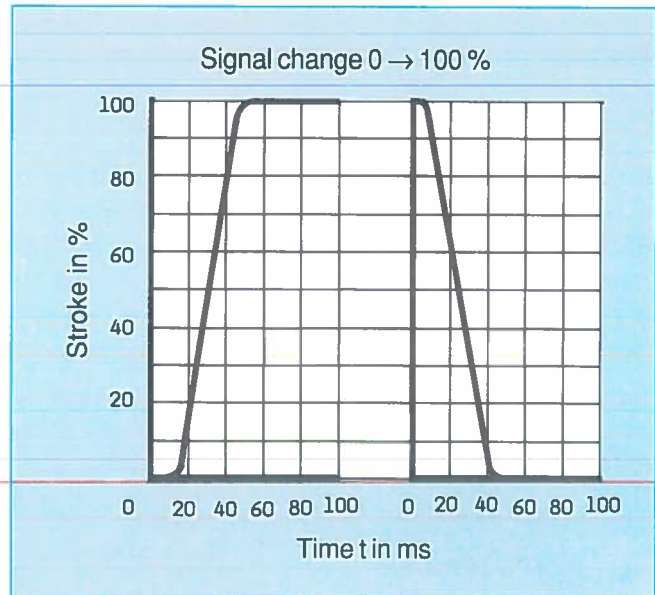
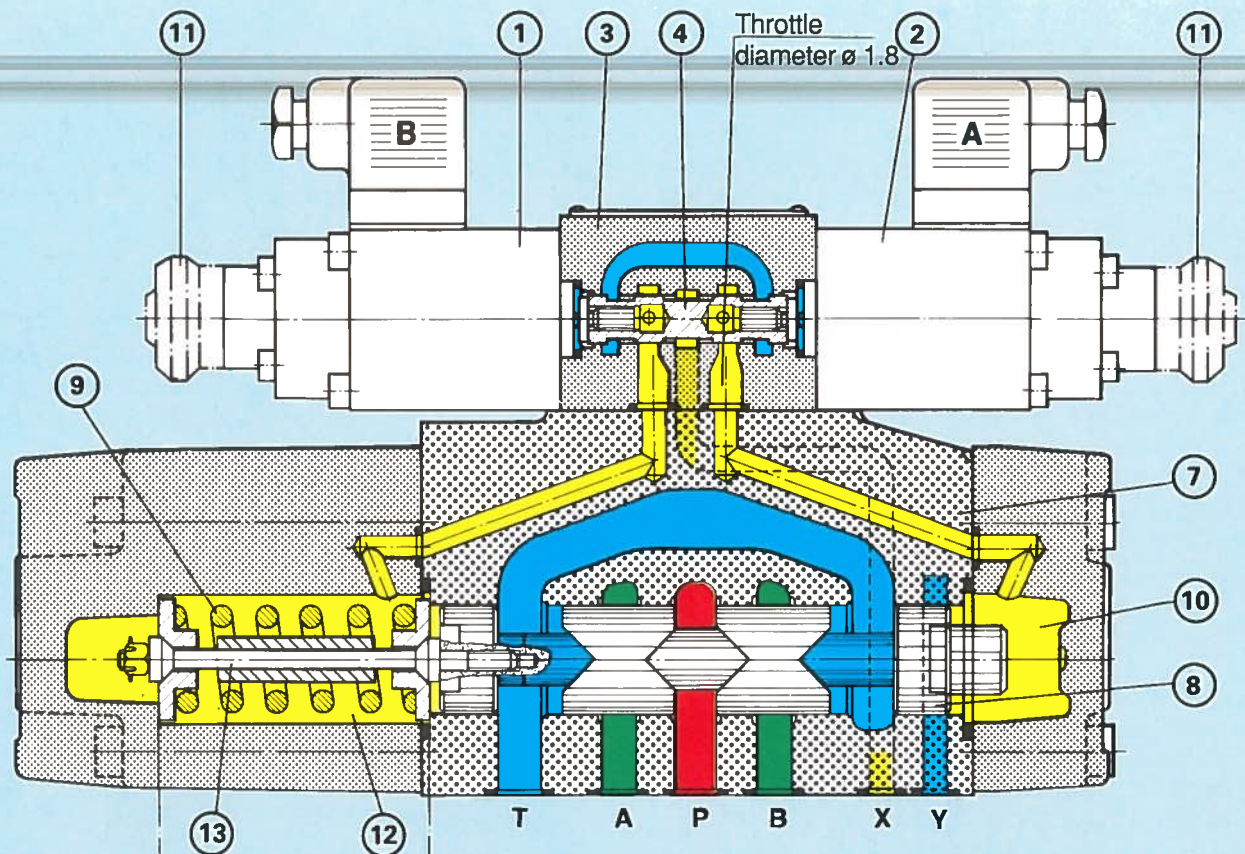


Fig. 32 Transition function for stepped electrical input signal



Type 4 WRZ 16 ...-30/6A...Z4

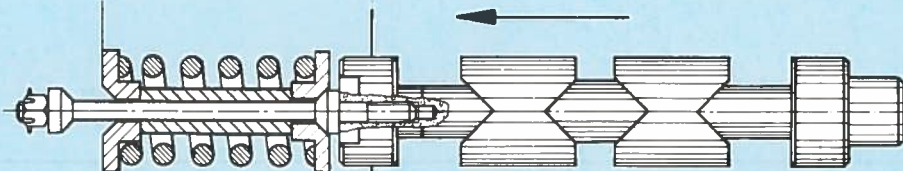


Fig. 33a

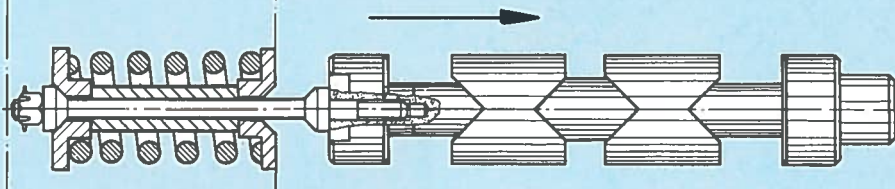
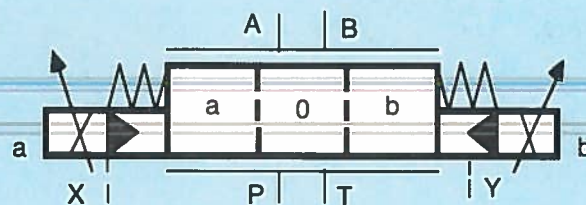


Fig. 33b

Symbol



X = external  
Y = external

Fig. 33 Pilot operated proportional directional valve Type 4 WRZ with single-sided "spring centering"

An explanation to the question which is often asked in this connection:

**"Is any one type of proportional valves with spool position feedback be preferred?"**

It is true that the reproducibility of the main spool position with electrical position feedback and constant oil temperature is within the range of 0.01 mm. However, the fact must be taken into consideration that at various oil temperatures (20 °C ... 70 °C), the temperature drift of the transducer and of the linkage can lead to changes in the spool position which were measured to be up to 0.03 ... 0.04 mm at the Mannesmann Rexroth laboratories, referred to a total stroke of 4 mm, by the spool of a proportional directional valve, Type 4 WRE 10. The reproducibility of the pilot operated proportional directional valves included in the Type 4 WRZ Mannesmann Rexroth range is in the region of 0.06 ... 0.07 mm. In this case, there is no temperature drift, but rather a direct spring feedback. The overall stroke is 5.5 mm.

The good repetition accuracy of the 4 WRZ valves is achieved by the high spring constant of the spring at the main spool in conjunction with the low friction spring centering (spherical dome) - large positioning forces relative to the possible disturbing forces.

Electrical feedback is considered necessary in the case of directly operated proportional directional valves since the relationship of the disturbing forces which occur to the available solenoid force is particularly unfavourable (relatively small solenoid force with respect to the disturbing forces).

The high precision, narrow shaped triangular grooves in the control spools play a significant role in the good reproducibility of the control process, both in the case of the directly operated and the pilot operated proportional directional valves included in the Rexroth range.

Mechanical friction, also caused by dirt particles in the oil, is of significance with regard to the repetition accuracy only when the same signal value is to be maintained over a long period of time - stick-slip effect. In view of the fast changes in the signal value, common to the majority of modern systems, the effect of the frictional value is extremely low. The valve spool is always operated above the stick-slip range.

In control processes, it is important that, in addition to good repetition accuracy and low hysteresis, the positioning device, i.e. the proportional directional valve also features good dynamics. However, this requirement can be met only incompletely with a proportional solenoid control system (inductive solenoid system). For this reason, a servo valve control system (torque motor) may also be recommended for these cases (see Fig. 37). The control characteristics of these devices with feed-back are improved with servo actuation.

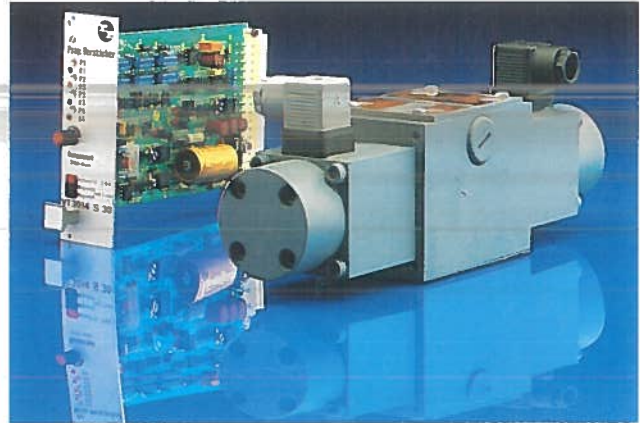


Fig. 34 Directly operated proportional directional valve without spool position feedback, Type 4 WRA 10, electronic controls

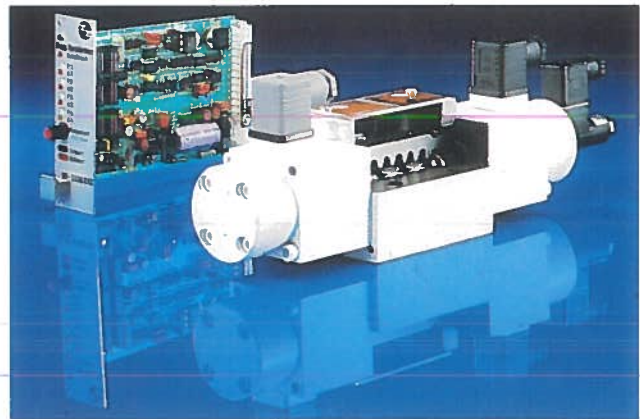


Fig. 35 Directly operated proportional directional valve with spool position feedback, Type 4 WRA 10, electronic controls

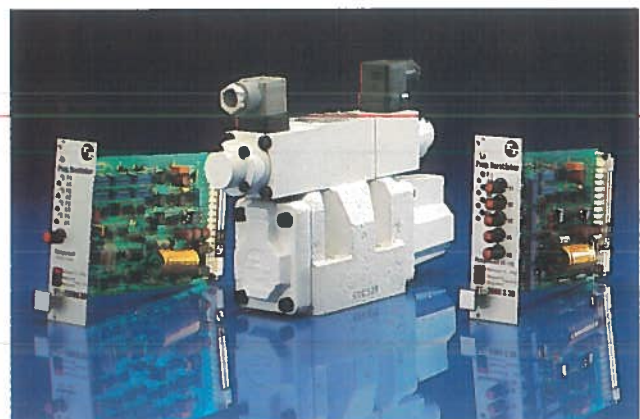


Fig. 36 Pilot operated proportional directional valve without spool position feedback, Type 4 WRZ 10, electronic controls

The advantage of pilot operated proportional directional valves without feedback lies in their uncomplicated design and the low requirements with regard to electronic circuitry, e.g. screened cables, laid separately to the positional transducer is rendered unnecessary.

It is thus not possible to make a clear cut decision for any one design on the subject of "Spool Position Feedback in Proportional Directional Valve". The best solution can be found only by taking into consideration the individual case and its requirements.

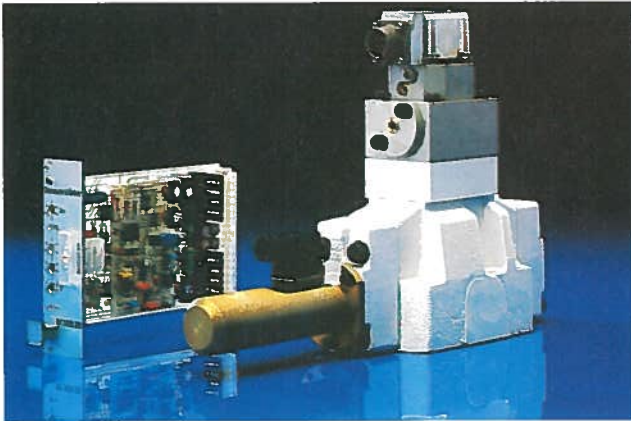


Fig. 37 Pilot operated 4-way control valve with spool position feedback, Type 4 WRD 16, electronic controls

To conclude the subject of proportional directional valves, the main features are itemized in the following:

1. Concept as 4/3-way valves with spring centered mid-position.
2. Low sensitivity to dirt.
3. Direction and flow control combined in one unit. With regard to program sequences, no additional directional control valves and throttles are necessary for rapid traverse and creep speeds. Speed transition not in steps but rather continuously variable.
4. Relatively long spool strokes as in normal directional valves.
5. The consumer is constantly controlled during meter-in and meter-out by two control lands.
6. In conjunction with electronic controls, acceleration and deceleration procedures can be realized extremely easily and reliably.

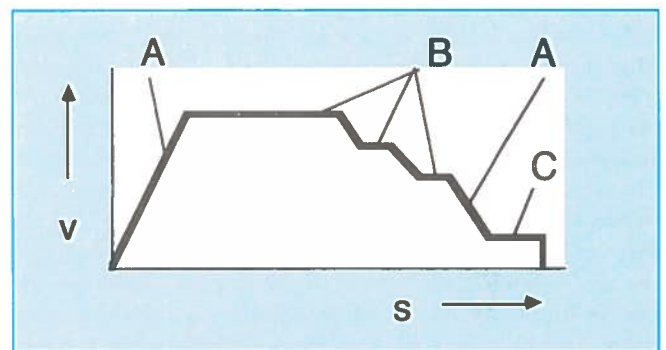


Fig. 38 Velocity/path diagram

- A = Acceleration or deceleration
- B = Various velocities
- C = Residual velocity before stop

Acceleration and deceleration periods are specified by the electronics and do not depend on hydraulic influences (oil viscosity).

7. Power consumption as for DC solenoids

## Proportional Pressure Control Valves

These valves are used for remote pressure setting by electrical means where pressure increase and pressure drop are additionally influenced with respect to time. In this way, the pressure can be changed, i.e. adapted, by means of an electrical signal value corresponding to process requirements.

### Direct Operated Proportional Pressure Relief Valve

The proportional pressure relief valve is designed as a poppet valve. It basically consists of the housing (1), proportional solenoid (2) with inductive positional transducer (3), valve seat (4), valve poppet (5) as well as compression spring (6) (Fig. 41).

The proportional solenoid is a position controlled solenoid. In this case, it replaces to a certain extent the manual setting by means of the adjusting spindle.

A signal value provided via the amplifier results in a stroke at the solenoid proportional to the signal value. This preloads the compression spring (6) via the spring plate (7) and presses the poppet against the seat. The position of the thrust pad (i.e. solenoid armature) and therefore indirectly the pressure setting is recorded by the inductive positional transducer and monitored by the electronic control system in a position control loop. Any control deviations from the signal value are corrected by the closed loop control system. Solenoid friction is thus compensated resulting in a high precision, reproducible pretensioning force of the spring: hysteresis 1 % referred to max. setting pressure, repetition accuracy 0.5 % referred to max. setting pressure.

The max. setting pressure depends on the pressure rating (25 bar, 180 bar, 315 bar). The various pressure ratings are achieved by different valve seats, i.e. with different seat diameters. Since the solenoid force remains constant, the highest pressure rating has the smallest diameter.

By way of example for the pressure rating 25 bar, it can be seen from the characteristic curves that the maximum setting pressure also depends on flow.

At signal value 0 - power failure to the proportional solenoid or cable breakage at positional transducer - the lowest setting pressure is assumed. (Dependent on the pressure rating and flow).

The spring (8) must also be mentioned in this connection. It ensures, at signal 0, that the moving parts, such as armature, are shifted back in order to always achieve the minimum value ( $p_{min}$ ). It also serves the purpose of compensating the weight of the armature when the valve is installed vertically.

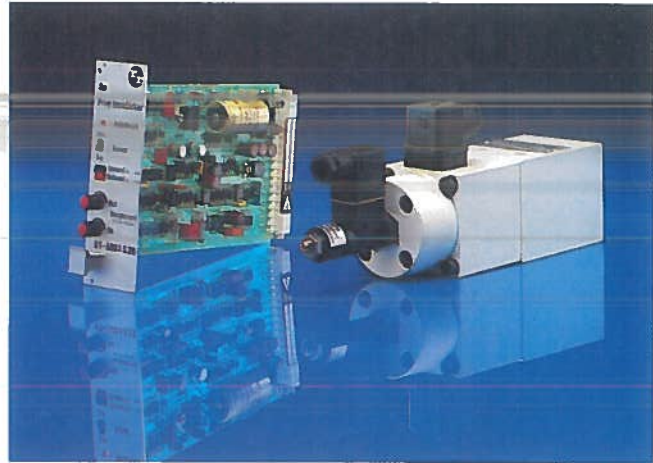


Fig. 39 Directly operated pressure relief valve Type DBETR, amplifier Type VT5003

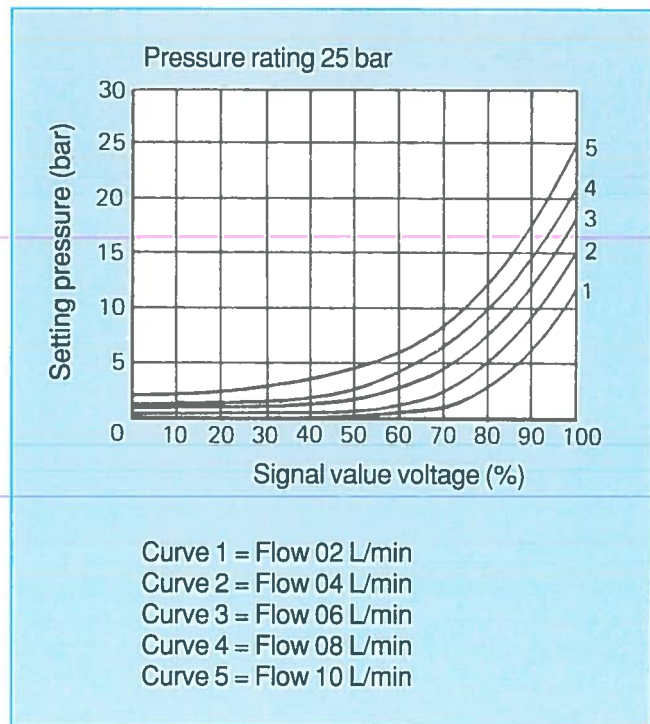
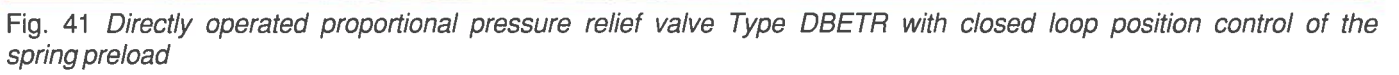


Fig. 40 Setting pressure dependent on the signal value voltage



## Pilot Controlled Proportional Pressure Relief Valve

Pilot controlled valves are used for applications involving higher flow rates.

The valve consists of the pilot valve (1) with proportional solenoid (2), optionally with integrated max. pressure safety control (3) and main valve (4) with main spool (5) (Fig. 44).

The basic function corresponds to that of the "normal" pilot controlled pressure relief valve. The difference lies in the pilot control section. In this case, the compression spring has been replaced by the proportional solenoid. It is a "force-controlled" proportional solenoid. A proportional force acting on the pilot poppet (6) therefore corresponds to a certain current value, specified by the electronic control. A higher input current signifies a greater solenoid force and therefore higher pressure setting; a lower input current signifies a lower pressure setting. The pressure applied by the system (port A) acts on the main spool (5). At the same time, the system pressure is applied to the spring-loaded side of the main spool (11) via the pilot line (10) equipped with the orifices (7,8,9). This system pressure acts on the pilot poppet (6) via the orifice (12) against the force of the proportional solenoid (2). The pilot poppet (6) opens when the system pressure increases above the set value corresponding to the solenoid force. The pilot oil can flow to the tank via the port Y (13). It should be noted that flow from port Y is always at zero pressure.

Due to the orifice combination in the pilot control line, a pressure drop now occurs at the main spool (5) such that it is raised from the seat and opens the connection from A to B (pump-tank).

To safeguard the system against impermissibly high currents at the proportional solenoid (2) which would inevitably result in impermissibly high pressures, a spring-loaded pressure relief valve can be installed as an additional option in the form of a max. pressure safety control (3). This valve can also be used to safeguard the pump.

When setting the pressure for the max. pressure safety control, a certain setting clearance with respect to the max. pressure setting must be maintained at the proportional solenoid to ensure that it really only responds at pressure peaks.

As a reference value, this "safety range" should be approx. 10 % of the max. operating pressure.

For example:

Max. operating pres. via electronic control = 100 bar,  
setting max. pressure safety control = 110 bar.

The various pressure ratings (e.g. 50, 100, 200, 315 bar) are once again achieved by means of different seat diameters. In addition to the standard characteris-

tic curves "Operating Pressure dependent on Flow" and "Minimum Setting Pressure dependent on Flow" the relationship between input pressure and power consumption is also of considerable importance.

The characteristic curve for the pressure rating 200 bar is provided as an example. The maximum pressure of a pressure rating is always achieved with the maximum current of 800 mA. For practical applications, this means that the pressure rating should be selected on the basis of the max. pressure required and no higher in order to obtain the best possible resolution.

The characteristic curve also shows that a higher hysteresis results when a different electrical control is used, e.g. not VT 2000 without dither current.

The following values can be achieved with this valve:

Linearity current - inlet pressure:	$\pm 3.5\%$
Repetition accuracy:	$< \pm 2\%$
Hysteresis:	$\pm 1.5\%$
Recommended filter element pore size:	$\leq 10\ \mu\text{m}$
(Pressure filter in supply line).	

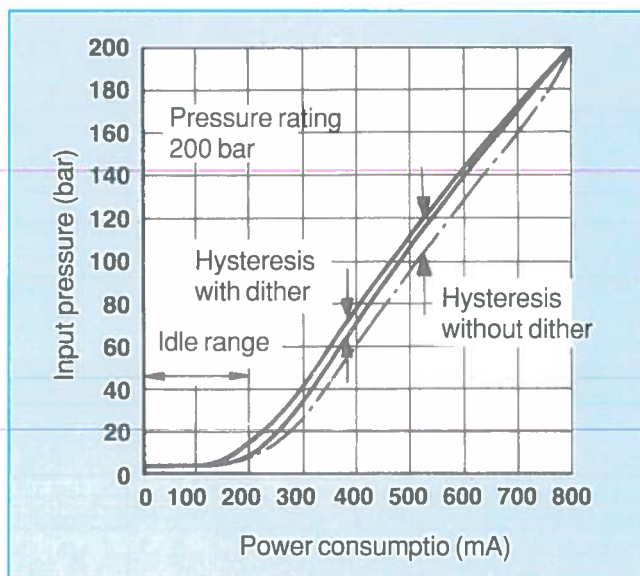


Fig. 42 Dependency of input pressure on power consumption

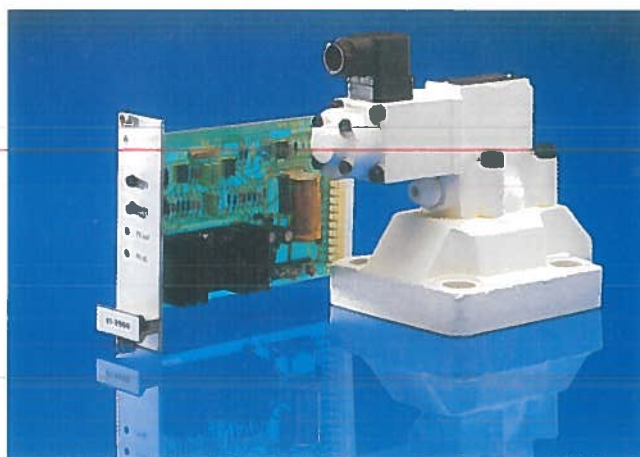


Fig. 43 Pilot operated proportional pressure relief valve Type DBE, electronic controls

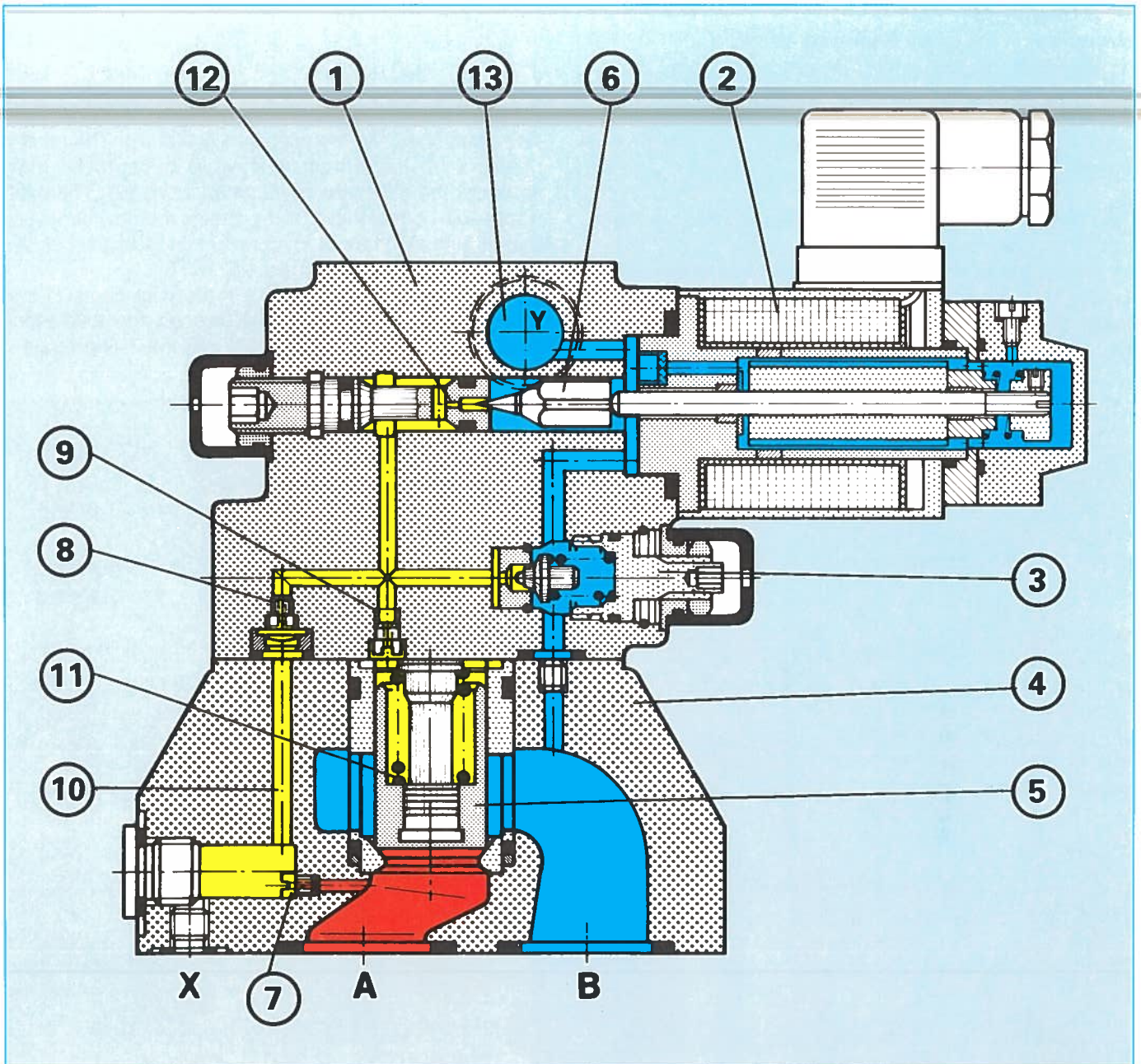
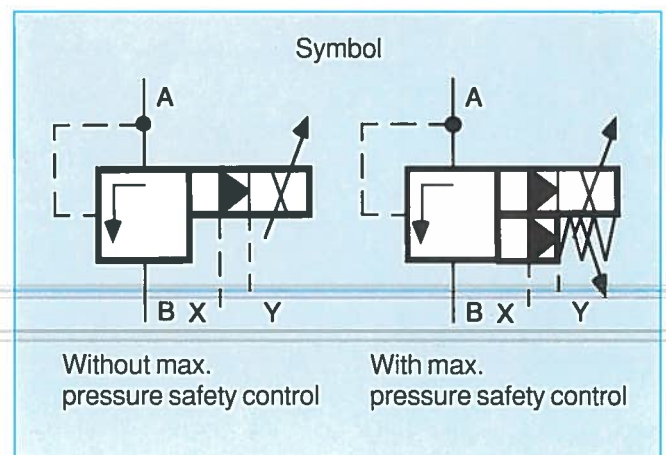


Fig. 44 Pilot operated proportional pressure relief valve with max. pressure safety control Type DBEM



The pressure is set in channel A dependent on current via the proportional solenoid (2).  
In the neutral position - signal value 0 (no pressure or flow at B) - the spring (10) holds the main spool in its initial position. The link from B to A is closed, so that a jump at the start is suppressed.

From the flow control valve (9), the pilot oil flows into the spring chamber (10) and through the holes (11) and (12) via the valve seat (13) into the Y-line (14, 15, 16) to the tank.



The pressure value required in channel A is set at the corresponding amplifier. The proportional solenoid forces the valve poppet (20) against the valve seat (13), thereby limiting the pressure in the spring chamber (10) to the set value. If the pressure in channel A is lower than the set signal value, then the higher pressure in the spring chamber (10) shifts the main spool to the right. The link from B to A is opened. There is equilibrium of forces at the main spool when the set pressure in A is reached.

Pressure in A • spool surface area (7) =  
pressure in spring chamber (10) • spool surface area +  
+ spring force (17)

If the pressure in A increases, the spool is shifted to the left in the closing direction B to A.

If the pressure in A is to be lowered in an static oil column (e.g. cylinder against limit stop), a lower pressure which is immediately applied in the spring chamber (10) is selected at the signal value potentiometer of the corresponding amplifier. The higher pressure in A acting on the surface area (7) of the main spool presses the main spool as far as it will go against the screw plug (18).

The connection from A to B is closed and from A to Y opened. The force of the spring (17) now acts against the hydraulic force on the area (7) of the main spool. In this position of the main spool, the pressure medium can flow from channel A via the control land (19) to Y and then to the tank.

When the pressure in A has dropped to the pressure in the spring chamber (10) plus  $\Delta p$  from spring (17), the main spool closes the large control holes in the bushing at the control land A to Y.

The residual pressure difference of approx. 10 bar with respect to the new set pressure in A is now relieved via the fine control hole (22). This arrangement ensures excellent settling characteristics without a dip-in pressure.

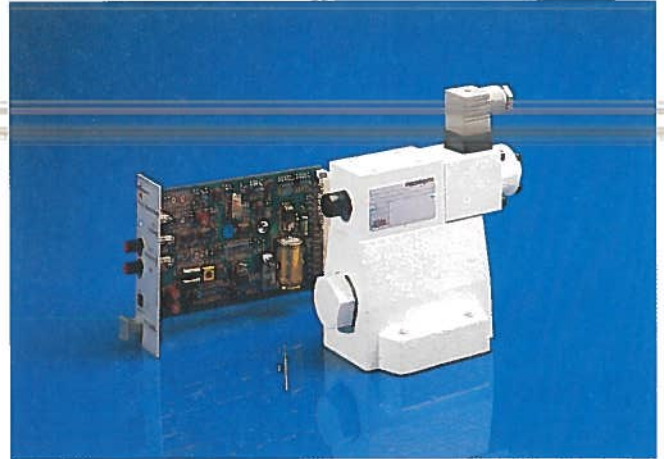


Fig. 46 Pilot operated proportional pressure reducing valve Type DREM 20, control electronics

A non-return valve (5) can be installed as an option to facilitate free return flow from channel A to B. A part of the flow of oil from channel A simultaneously flows via the open control land (19) of the main spool from A to Y and to the tank.

### Type DREM

A spring-loaded max. pressure relief valve (21) can be installed on request to provide a hydraulic safeguard against impermissibly high electrical control current at the proportional solenoid which inevitably results in high pressures in port A.

**Note:** When the hydraulic oil flows back from channel A to channel B via the non-return valve (5), the simultaneous parallel flow via Y to the tank influences deceleration of the consumer connected to A when deceleration is facilitated by a throttle valve (e.g. proportional directional valve) in channel B. The third path A to Y is not suitable for use as a pressure relief valve in channel A.

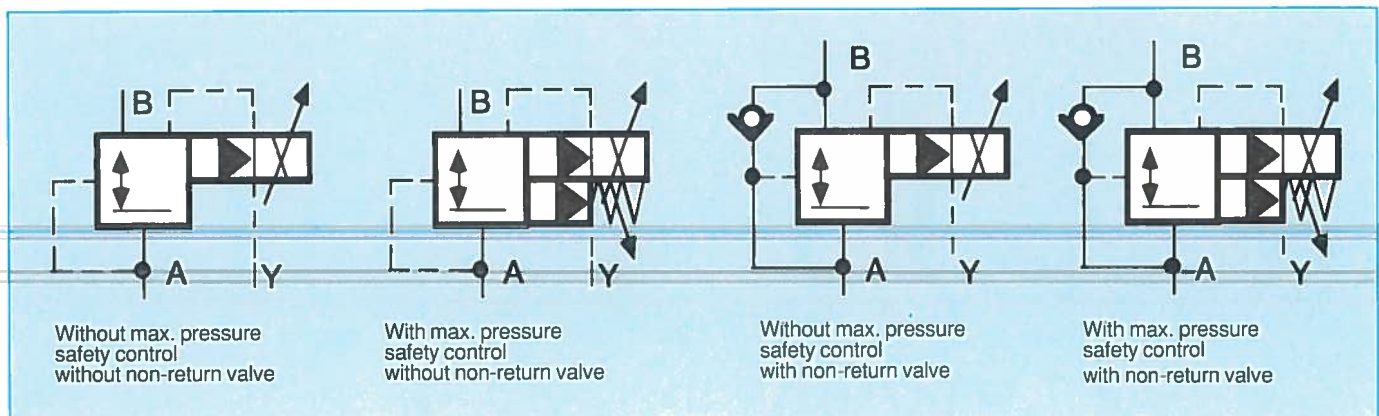


Fig. 47 Symbols

## Pilot Operated Proportional Pressure Reducing Valve, Type DRE 30

The pressure in channel A is set current-dependent via a proportional solenoid.

In neutral position - no pressure in channel B - the main spool (4) is opened from channel B to channel A.

The pressure in channel A acts on the underside of the main spool in the closing direction, while the pressure of the pilot valve acts on the spring side of the main spool in the opening direction from channel B to A.

The pilot oil is obtained from channel B and flows via the hole (6), fixed flow control (9), hole (7), valve seat (10) passed the valve poppet (8), via the Y channel to the tank.

Dependent on the electrical signal value at the proportional solenoid (2), a pressure is built up at the pilot valve (1) which acts on the spring side of the main spool. In the controlled position of the main spool (4), the oil flows from channel B to A such that the pressure in channel A is not exceeded (setting of the pilot valve plus main spool spring).

If the consumer connected to port A does not move (e.g. cylinder piston against limit stop) and a lower pres-

sure is set via the proportional solenoid (2) for channel A, then the main spool (4) closes the connection from channel B to A and at the same time opens the connection from channel A to the spring chamber of the main spool (4). In this position, the compression volume in channel A can be relieved via the pilot valve (1) and the port Y.

A non-return valve (11) can be installed as an option to facilitate free return flow from channel A to B.

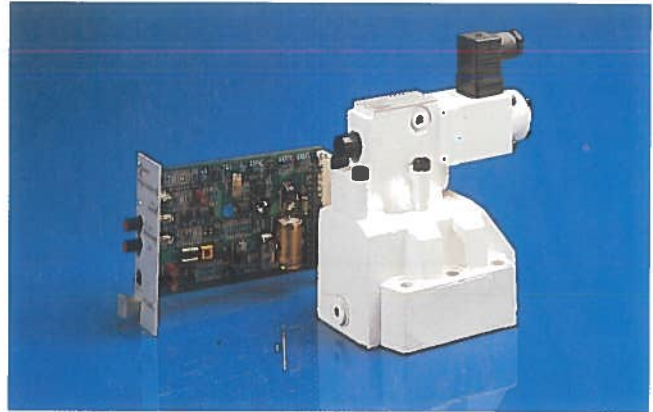


Fig. 48 Pilot operated proportional pressure reducing valve Type DRE 30, electronic controls

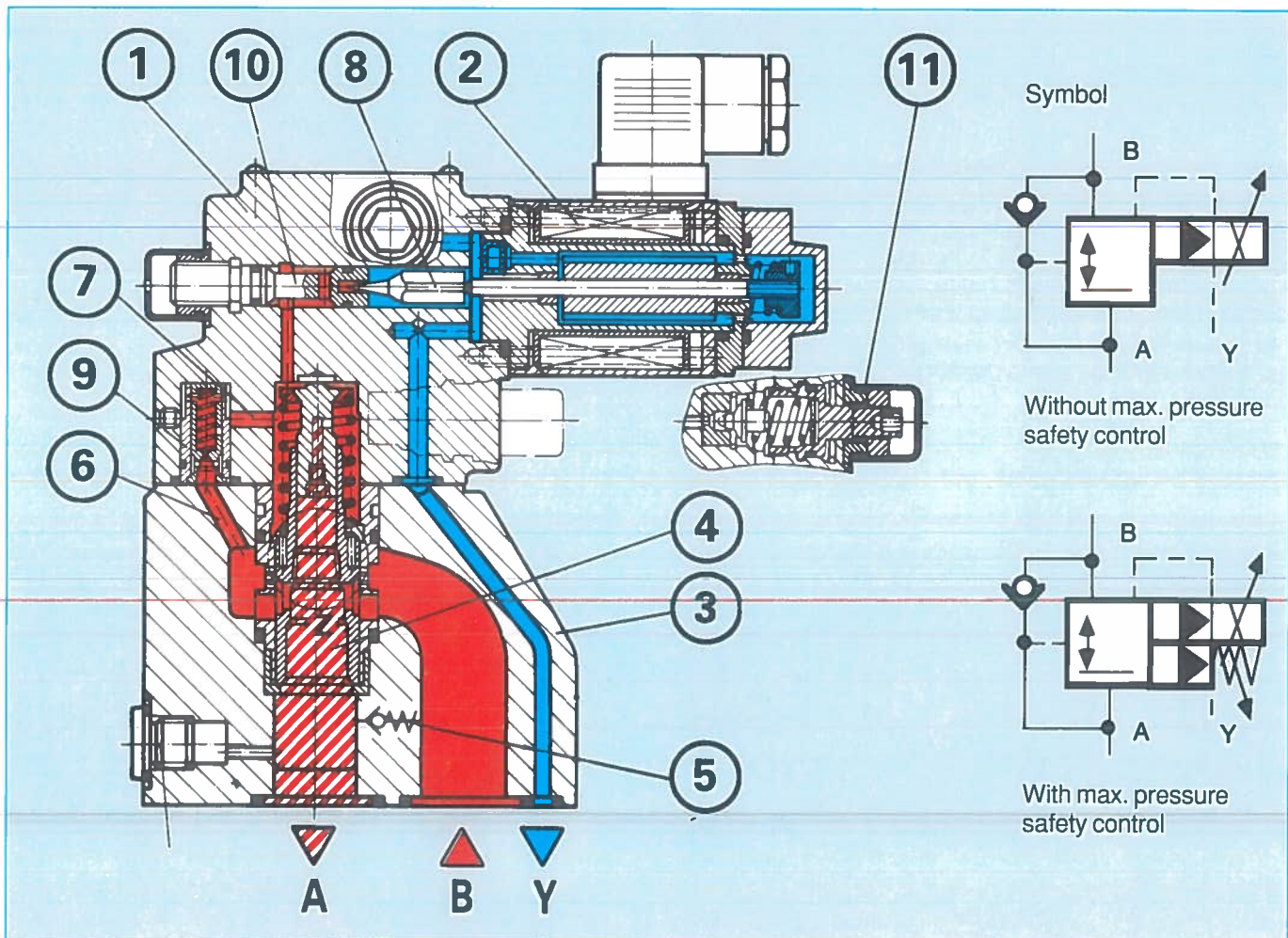


Fig. 49 Pilot operated proportional pressure reducing valve Type DRE 30/DREM 30

## Proportional Flow Control Valves

### 2-Way Proportional Flow Control Valve with Downstream Pressure Compensator (Size 6)

The 2-way proportional flow control valve can control, independent of pressure and temperature, oil flow specified by the electrical signal value. The most important components are the housing (1), the proportional solenoid with inductive positional transducer (2), the measuring orifice (3), the pressure compensator (4) and the (optionally) installed non-return valve (5).

The oil flow setting is determined by an electrical signal (signal value) set at a potentiometer. In conjunction with the electronic control (e.g. amplifier Type VT 5010), this set signal value results in a corresponding current and therefore a proportional stroke of the proportional solenoid (stroke-controlled solenoid). Correspondingly, the measuring orifice (3) is shifted downwards, thereby releasing an opening to flow. The position of the measuring orifice is fed back by the inductive positional transducer. Any deviations from the signal value are corrected by the closed loop control. The pressure compensator maintains the pressure drop at the measuring orifice at a constant value. The oil flow is therefore independent of load. Good design of the measuring orifice ensures a low temperature drift.

The measuring orifice is closed when the signal value is zero. The measuring orifice closes in the case of power failure or cable breakage at the electrical positional transducer.



Fig. 50 2-Way proportional flow control valve Type 2 FRE 6, electronic controls

Starting without jump is possible from zero signal. The measuring orifice can be opened and closed with a delay via two ramps in the electrical amplifier.

Free return flow from B to A is possible via the non-return valve (5).

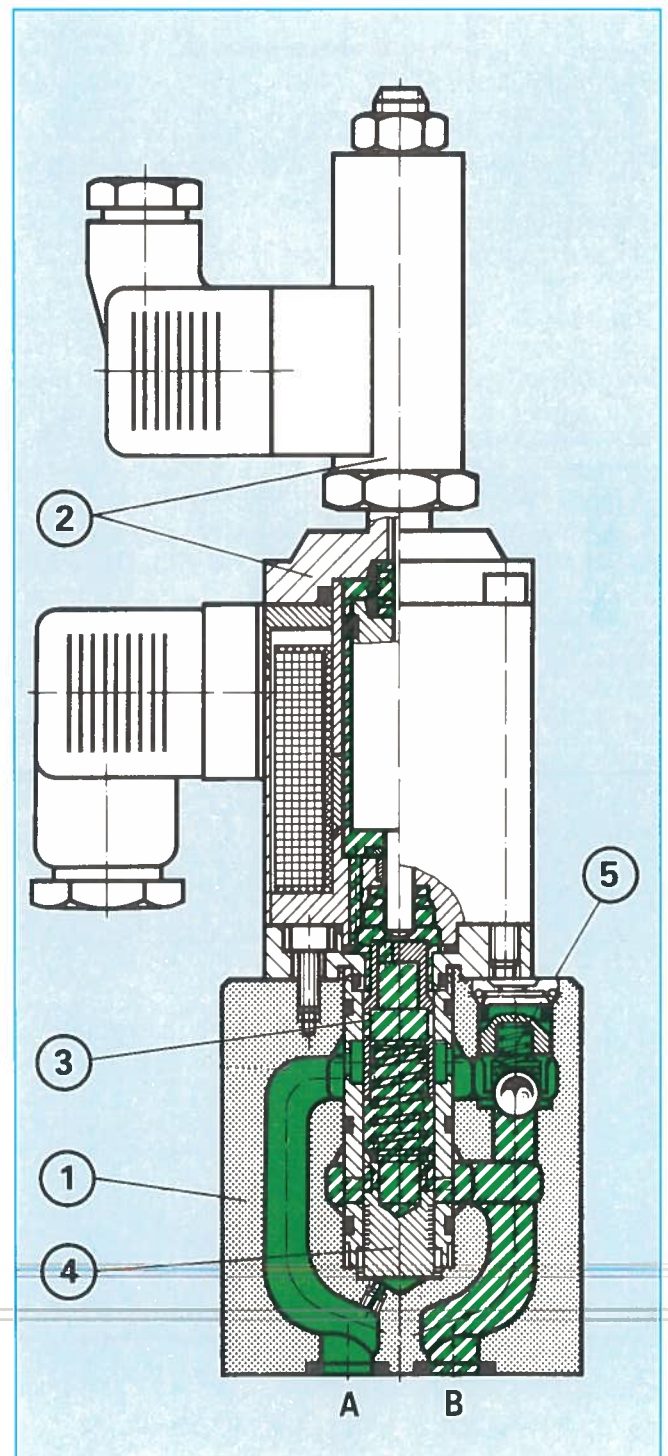


Fig. 51 Proportional flow control valve Type 2 FRE 6

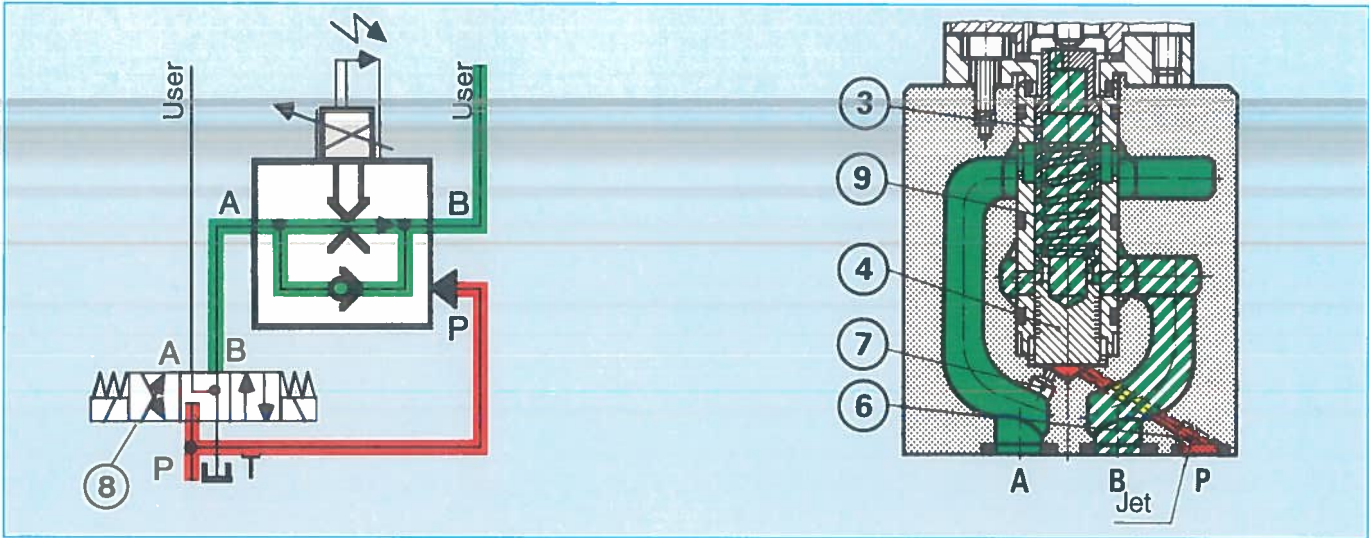


Fig. 52 External pilot operation of pressure compensator

### External closing of the Pressure Compensator

The basic function corresponds to those of the 2-way proportional flow control valve already described. However, the pressure compensator (4) is closed with pressure applied via port B (6) (Fig. 52) to facilitate suppression of a jump when starting with the measuring orifice (3) open (signal value > 0). The internal link (7) between port A and the effective area of the pressure compensator (4) is closed. In this way, the pressure in P prior to the directional valve (8) (see circuit example) acts via the external port P (6) on the pressure compensator (4), thereby holding it against the force of the spring (9) in the closed position. If the directional control valve (8) is shifted to the left spool position (connection P→B), then the pressure compensator (4) moves

from the closed position to the controlled position so that the jump at start is prevented.

Through the use of different measuring orifices, various maximum flows can be achieved at 100 % signal value. The characteristic curves in Fig. 53 illustrate the variations.

A fine control range of, for example, up to 2 l/min (Fig. 54) can be achieved with a correspondingly designed orifice opening. The electrical signal value can be infinitely varied between 0 and maximum. The frequency response (for explanation of the term frequency response refer to "Introduction to Servo Valve Technology") indicates the rapidity of the valve (Fig. 55).

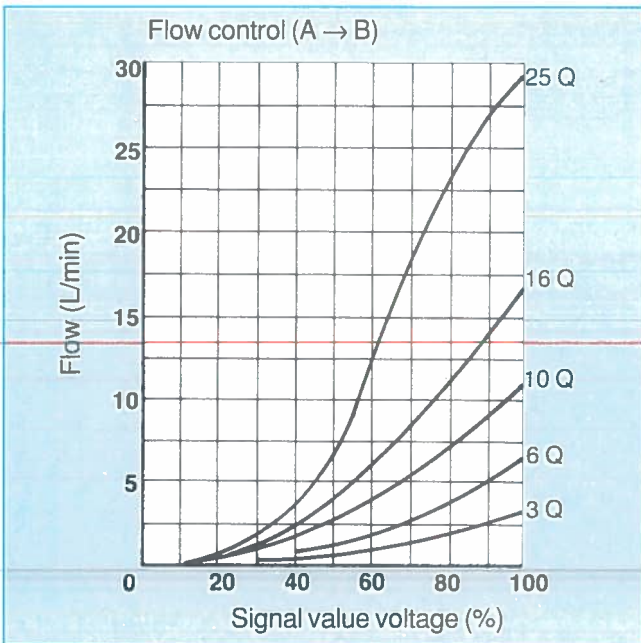


Fig. 53 Flow as function of signal value voltage

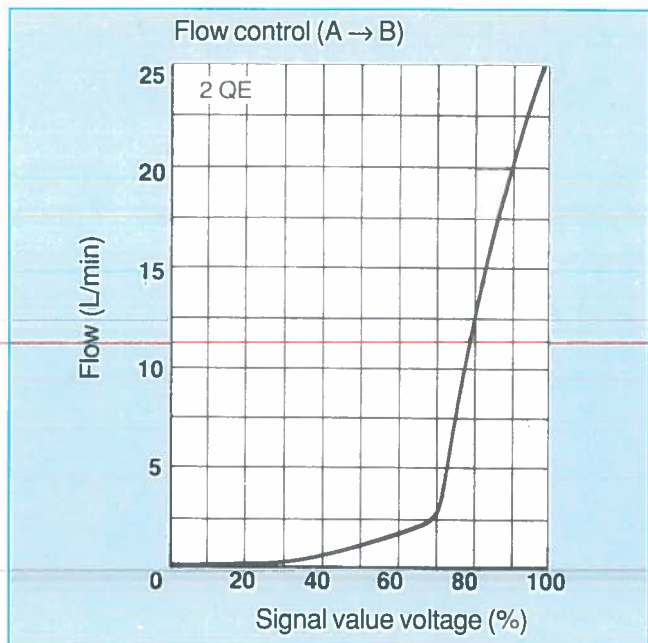
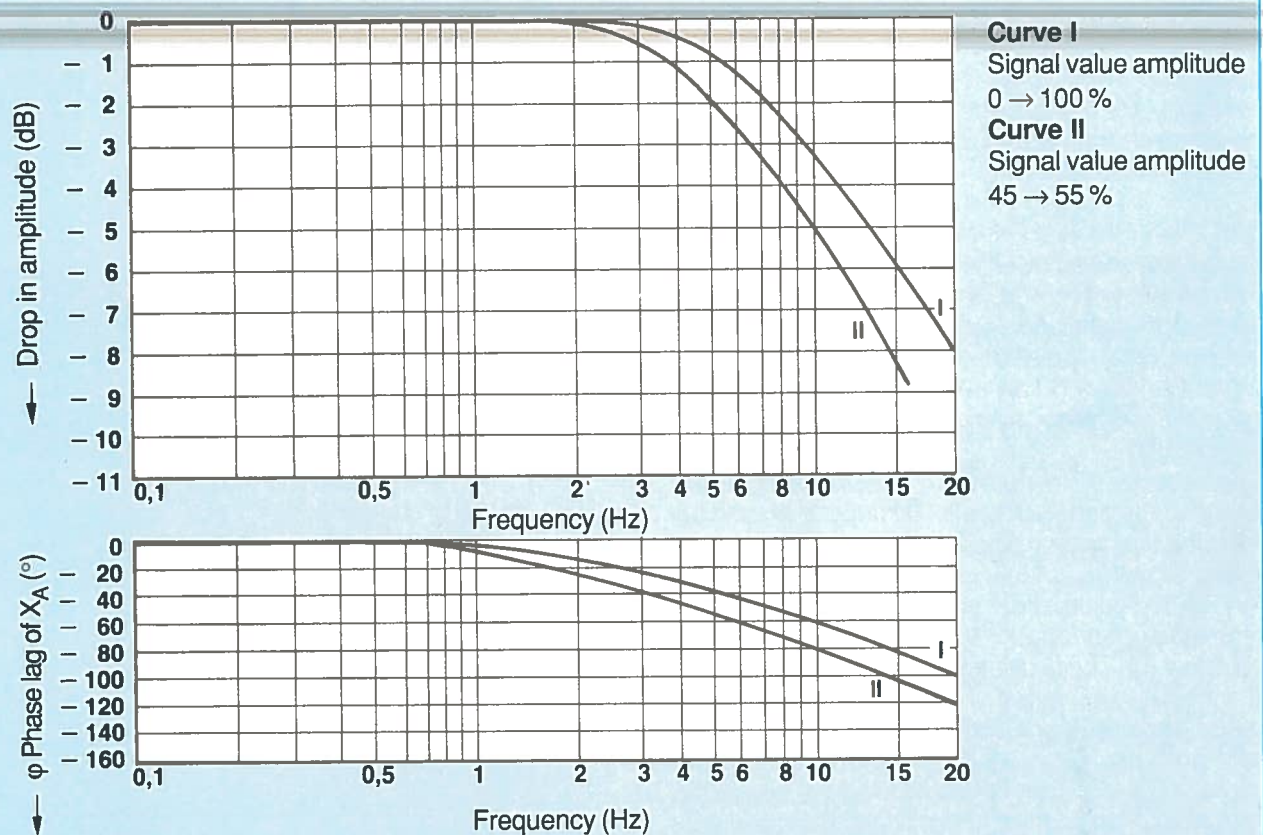


Fig. 54 Flow as function of signal value voltage for valves with progressive characteristics and stepped rapid traverse



Dynamics-table

Step response (%)	$Q_{\min}$ to $Q_{\max}$ $T_u + T_g$ (ms)	$Q_{\max}$ to $Q_{\min}$ $T_u + T_g$ (ms)
0 → 100	50	60
10 → 90	45	50
25 → 75	40	45

 $T_u$  = Time delay $T_g$  = Compensation time

Fig. 55 Frequency response

### 2-Way Proportional Flow Control Valve with Upstream Pressure Compensator (Size 10 and 16)

This type of valve is also described in the following for the sake of completeness. "Only" for the sake of completeness, not because it is of little significance but rather because the electrical signal conversion and the hydraulic section are well known. The opening to flow is changed by means of the stroke of the stroke-controlled proportional solenoid. The flow control function is obtained by the interaction of the throttle orifice and pressure compensator.

The characteristic flow curves can be linear or progressive depending on the shape of the orifice.



Fig. 56 2-Way proportional flow control valve Type 2 FFE 10, electronic controls

## 2-Way Proportional Throttle Valve (Cartridge valve)

This combination unit, can be used as a throttle (orifice) or in conjunction with a pressure compensator for controlling high flows. The applications include, for example, control systems for presses and plastic processing machines. Despite the high flow rates, the unit has a fast response time.

The 2-way throttle valve is an orifice with its opening stroke determined by an electrical signal value.

The throttle valve is supplied as a unit ready for installation with installation dimensions to DIN 24 342. The bushing (2) is screwed into the cover (1) together with the orifice spool (3) as well as the positional transducer (4) and the pilot control (5), including proportional solenoid (6).

The direction of flow is from A to B. The pilot oil port X is linked to the port A. The pilot oil outlet Y should be routed to the tank at zero pressure.

At signal value 0 (no current applied to proportional solenoid (6)) the pressure in port A acts via the pilot line X and the control spool (10) in addition to the spring in chamber (8). The orifice spool (3) is held closed.

If a signal value is fed to the amplifier card, the signal value (external signal) is compared in the amplifier (7) with the actual value (feedback of the transducer signal). The proportional solenoid (6) is energized with a current corresponding to the differential value.

The solenoid shifts the spool (10) against the spring (11). As the result of interaction between the throttling points (13) and (14), the pressure in the spring chamber (8) is set such that the spring-loaded orifice spool (3) assumes a position corresponding to the preset signal value and therefore determines the flow.

The orifice spool closes automatically (for safety) in the case of power failure or cable breakage. The components of the position control loop are designed such that the signal value and the stroke of the orifice spool (3) are directly proportional with respect to each other. Consequently, for constant pressure differences at the orifice, the volumetric flow from A to B is only dependent on the stroke of the orifice spool and the window geometry (9).

Direct proportionality between the signal value and volumetric flow is applicable to the system with linear opening law (FE..C10/L). The quadratic opening law (version FE..C10/Q) signifies a volumetric flow increasing as the (signal value)<sup>2</sup>.

The two characteristic curves illustrate this fact.

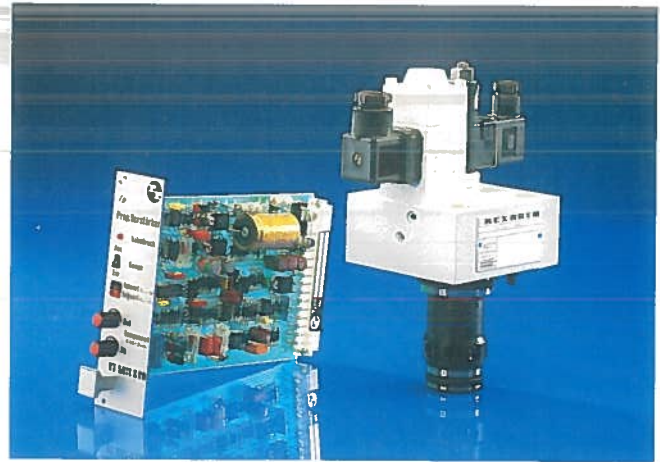


Fig. 57 2-Way proportional throttle valve Type FE..C, electronic controls



Fig. 58 Progressive flow characteristics

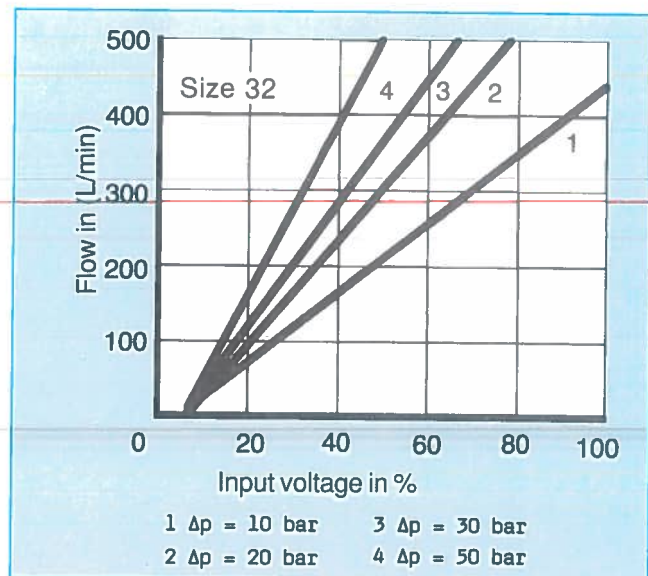


Fig. 59 Linear flow characteristics

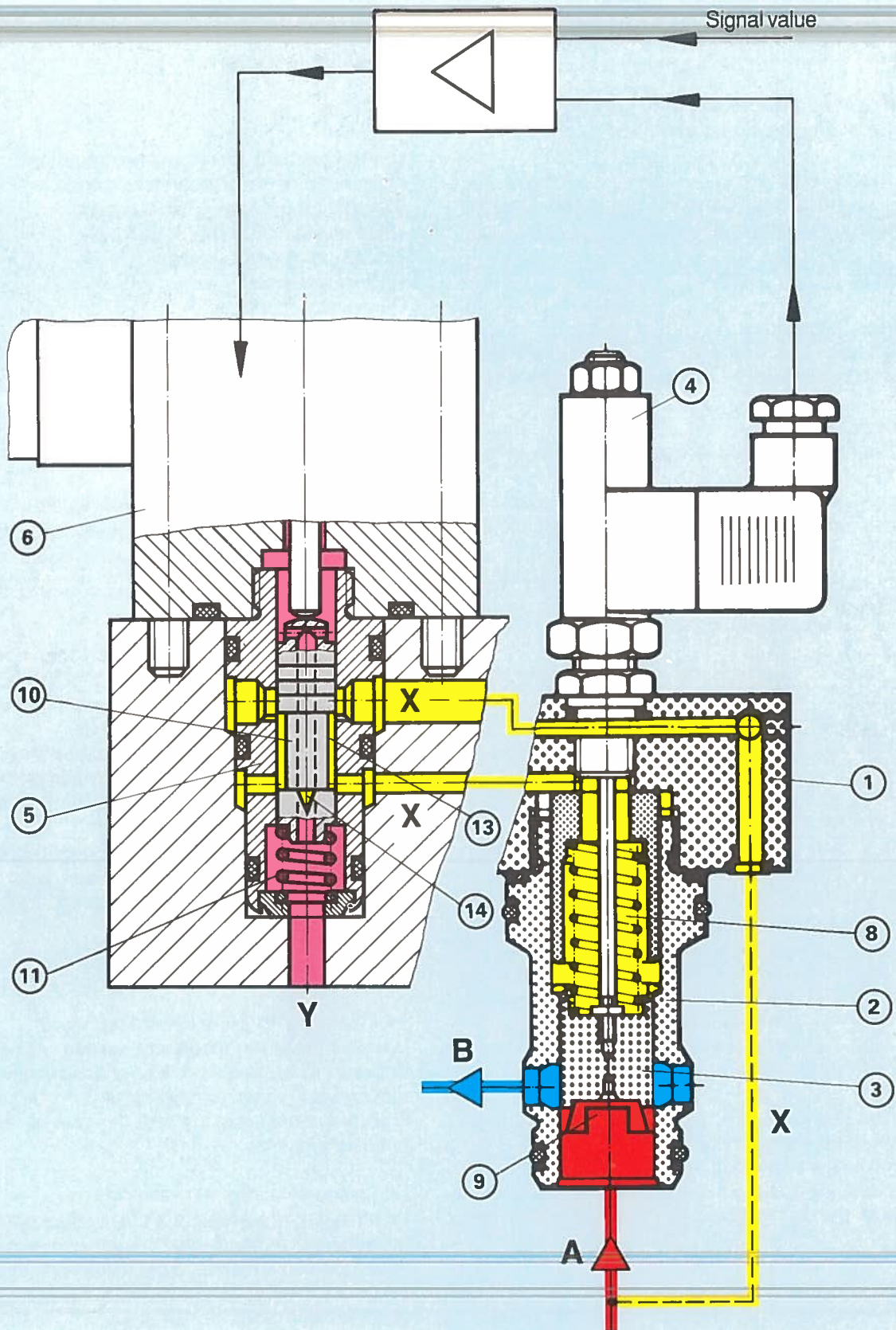


Fig. 60 2-Way proportional throttle valve (cartridge valve) Type FE..C

# Installation, Commissioning and Maintenance of Hydraulic Proportional Valves

## 1. General

In addition to the notes below, advice given in the following publications should be observed to ensure correct operation:

- the data sheets
- the VDI Specifications, Commissioning and Maintenance of Hydraulic Systems, VDI (3027)

## 2. Installation

### 2.1 Installation Instructions

Before the valve is mounted in the system, the valve type designation should be compared with the order data.

#### 1. Cleanliness:

- of surrounding area and proportional valve when mounting the unit.
- the tank must be sealed from external contamination
- prior to installation, pipes and the tank must be cleaned of dirt, scale, sand, metal chips etc.
- hot bent, or welded pipes must be subsequently pickled, flushed and lubricated
- only use fluff-free material or special paper for cleaning purposes

2. Sealing material such as hemp, putty or sealing tape must not be used.

3. To ensure high stiffness of the system, hose lines should not be used between the valve and the driven unit.

4. The piping system must be made up of seamless precision steel pipes in accordance with DIN 2391/C.

5. The connecting lines between the driven unit and valve must be as short as possible;

We advise that the proportional valve is installed in as near as possible to the driven unit.

The mounting surface must have a surface finish of  $Rt_{max} \leq 4 \mu m$  and be flat to 0.01 mm/100 mm length.

6. The mounting screws must as specified in the data sheet and tightened to the specified torque.

7. An oil bath air filter is recommended as the filter breather. Mesh width  $\leq 60 \mu m$ .

### 2.2 Installation Position

Any position, preferably horizontal; however, if the proportional valve is mounted on a driven unit, care should be taken to ensure that the valve spool is not positioned parallel to the acceleration direction of the driven unit.

### 2.3 Electrical Connection

Refer to the relevant data sheet for electrical connections.

Special protection classes require special measures which are stipulated on the relevant data sheet.

## 3. Commissioning

### 3.1 Fluids

Note recommendations given in data sheet.

Note pressure and temperature ranges.

Generally, the following can be used:

- mineral oil H-LP to DIN 51 525
- polyglycol in water solution
- phosphate-ester

Other fluids on request.

The maximum temperatures recommended by the manufacturer of the fluid should, if possible, not be exceeded in order to protect the pressure medium. To ensure constant response characteristics of the system, it is advisable to maintain the oil temperature constant ( $\pm 5^\circ C$ ).

### 3.2 Is the Correct Sealing Material Used?

When using HFD (phosphate ester) fluids, and also for high temperatures  $< 90^\circ C$ , viton seals (designation "V") must be used.

### 3.3 Filtration

- To ensure long service life, use 10  $\mu m$  absolute pressure line filters for the proportional control, the filter element pore size specified in the data sheet can however also be used.

- The permissible differential pressure for pressure line filters must be greater than the operating pressure.

- We recommend filters equipped with a filter clogging indicator.

- Absolute cleanliness must be ensured during filter change. Contamination at the outlet side of the filter is flushed into the system and causes faults and malfunctions.

Contamination at the inlet side reduces the service life of the filter element.

### 3.4 Operating Pressure for the Pilot Valve

The pilot pressure should not exceed 30 bar. If the pilot pressure exceeds 100 bar, a sandwich plate pressure reducing valve must be installed on the inlet side.

Pressure peaks from the tank line can be avoided with a non-return valve.

### 3.5 Solenoid Venting

To ensure perfect operation, it is necessary to bleed the solenoid at the highest point of the valve during initial operation. Under certain installation conditions, the tank line must be prevented from running empty by the installation of a preload valve.

## **4. Maintenance**

### **4.1 Return of Valve for Servicing**

When returning a defective valve, it is necessary to protect the base surface of the valve from the effects of dirt.

Careful and adequate packing is advisable to ensure no further damage is incurred during transport.

## **5. Storage**

Storage room requirements:

- dry, dust-free room, free of corrosive materials and vapours

For storage longer than 3 months:

- fill housing with preservative oil and seal.

Notes

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# **Load Compensation with Pressure Compensators**

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Dieter Kretz



## Pressure Compensators

All proportional directional valves which have been described up until now only represent throttle valves, in which the volumetric flow also changes as the pressure ratio changes. The volumetric flow decreases as the load pressure at the driven unit increases, conversely the volumetric flow increases as the load pressure decreases. Throttle valves are therefore useful as control devices only when the loads do not vary widely.

Fig. 1 shows a typical characteristic throttle curve. The change in volumetric flow is clearly shown dependent on the drop in valve pressure which is directly dependent on the load pressure in the case of constant pump and tank pressure.

$$p_V = p_S - \Delta p_L - \Delta p_T$$

$p_V$  = Valve pressure drop

$p_S$  = System pressure = constant

$\Delta p_T$  = Tank pressure = constant

$\Delta p_L$  = Load pressure = variable

It is therefore necessary to compensate the load influences described above by means of suitable devices.

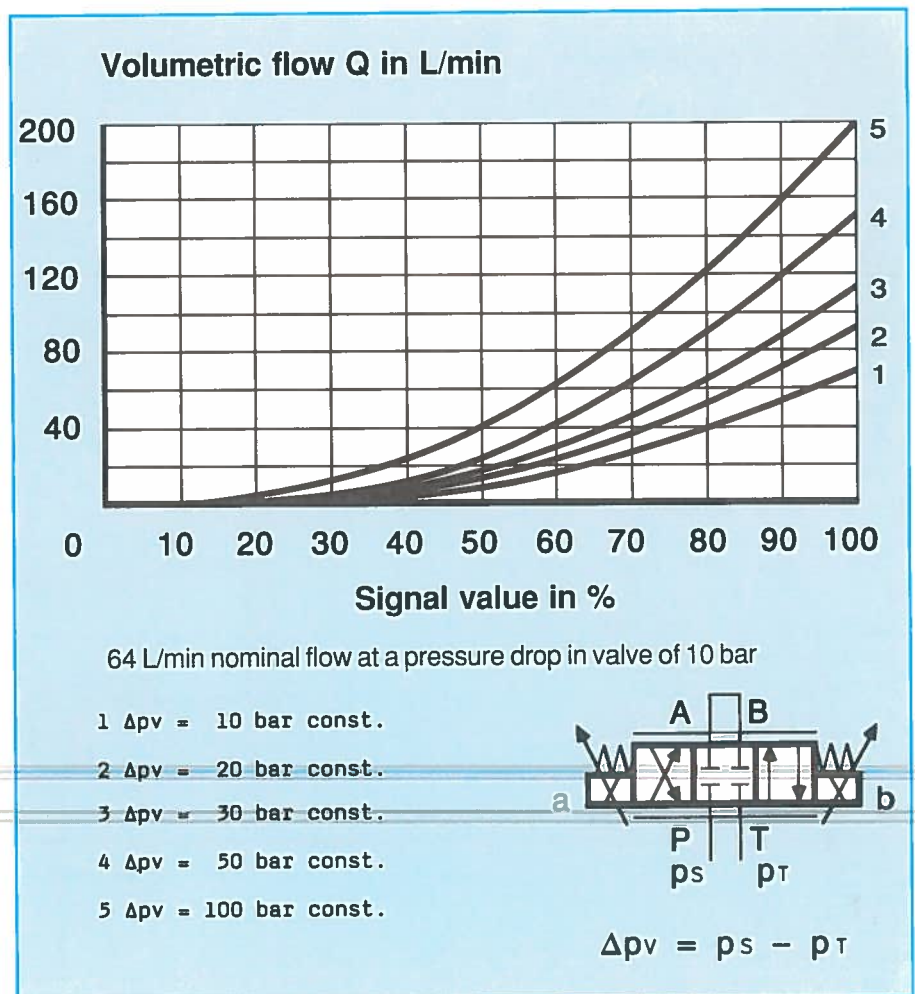


Fig. 1 Characteristic throttle curves of a proportional directional valve

## Load Compensation with 2-way Meter-in Pressure Compensator

The use of a 2-way meter-in pressure compensator, see Fig. 2, ensures the pressure drop at the meter-in throttle edge of the proportional valve is maintained constant.

In this way, load pressure fluctuations and changes in pump pressure are compensated. This also means that the flow cannot be increased by increasing the pump pressure. The valve must therefore be selected with regard to its nominal flow in accordance with the differential control pressure of the pressure compensator.

### Function of the 2-way Meter-in Pressure Compensator

The control orifice  $A_1$  and the measuring orifice  $A_2$  are arranged in the 2-way meter-in pressure compensator one after the other. Referred to the balanced position of the spool, it will be shown that the pressure drop  $\Delta p = p_1 - p_2$  at the measuring orifice remains constant as the consumer pressure varies. Without taking the flow force into consideration, the following is applicable for the balanced position

$$p_1 \cdot A_K = p_2 \cdot A_K + F_F$$

resulting in

$$\Delta p = p_1 - p_2 = F_F / A_K \approx \text{constant}$$

Since a light spring is installed and the control stroke is short, the change in the spring force is only slight and therefore the pressure drop almost constant. The control spool can only change the opening of the control orifice  $A_1$  when the spring force has been overcome. The flow control function is therefore effective only when the outer pressure difference  $p_P - p_2$  is greater than  $F_F / A_K$  (control -  $\Delta p$ ).

If the resistance to flow increases as the flow increases, then the outer pressure difference must also increase in order to achieve the flow control function.

Fig. 3 Principle diagram 2-way meter-in pressure compensator

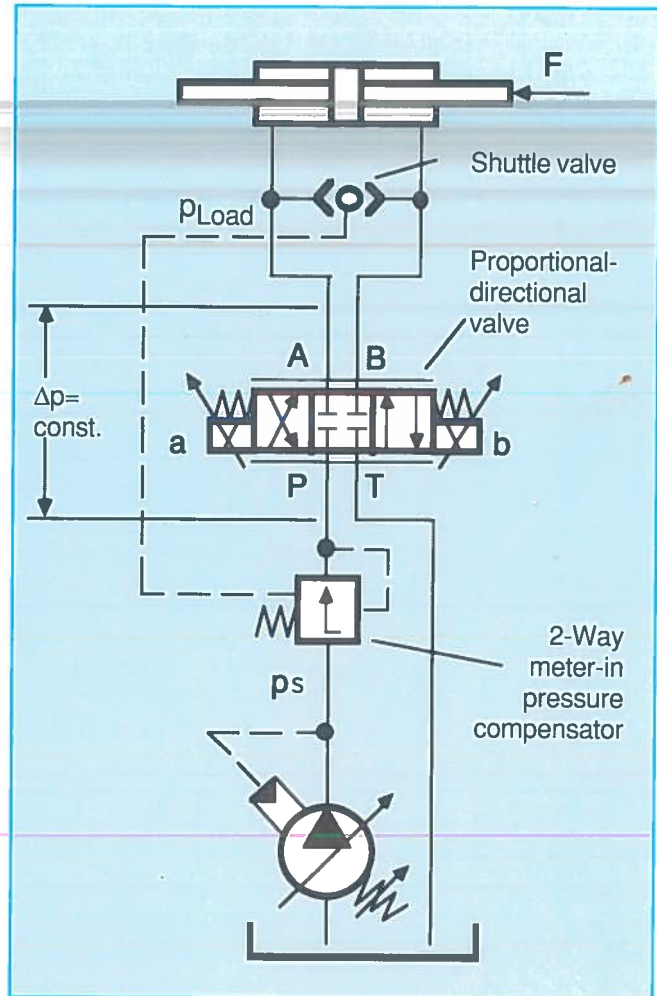
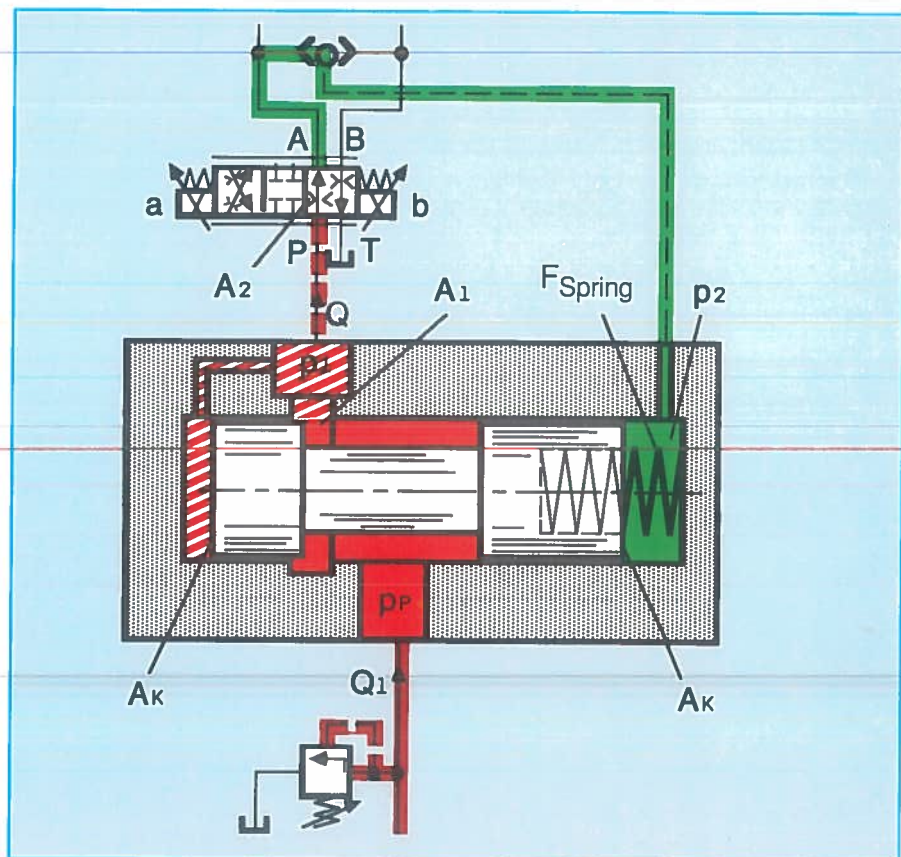


Fig. 2 Circuit example



## 2-way Meter-in Pressure Compensator in the P port, Type ZDC (Sandwich plate design)

Type ZDC 10 valves are directly operating sandwich plate valves, either 2-way or 3-way units.

They are used as meter-in pressure compensators for load compensation in the P port of the main valve.

The valve basically consists of the housing (1), the control spool (2), compression spring (3) with thrust pad (4) and the cover (5) with built-in shuttle valve (6).

The compression spring (3) holds the control spool (2) in the open position from P to P1 whenever the pressure difference  $P1 \rightarrow$  or  $P1 \rightarrow B$  is less than 10 bar. If the pressure difference exceeds 10 bar, the spool is moved to the left until the differential pressure is restored once more.

The signal and pilot oil are both fed internally from the pilot line (7) from channel P1. The pilot oil required (X-channel) for the pilot operated proportional directional valve (4 WRZ) can either be obtained internally from the P channel or externally as required.

The 3-way pressure compensator differs only with regard to the design of the spool.

The 2-way and 3-way pressure compensator is available in the sizes 10, 16 and 25.

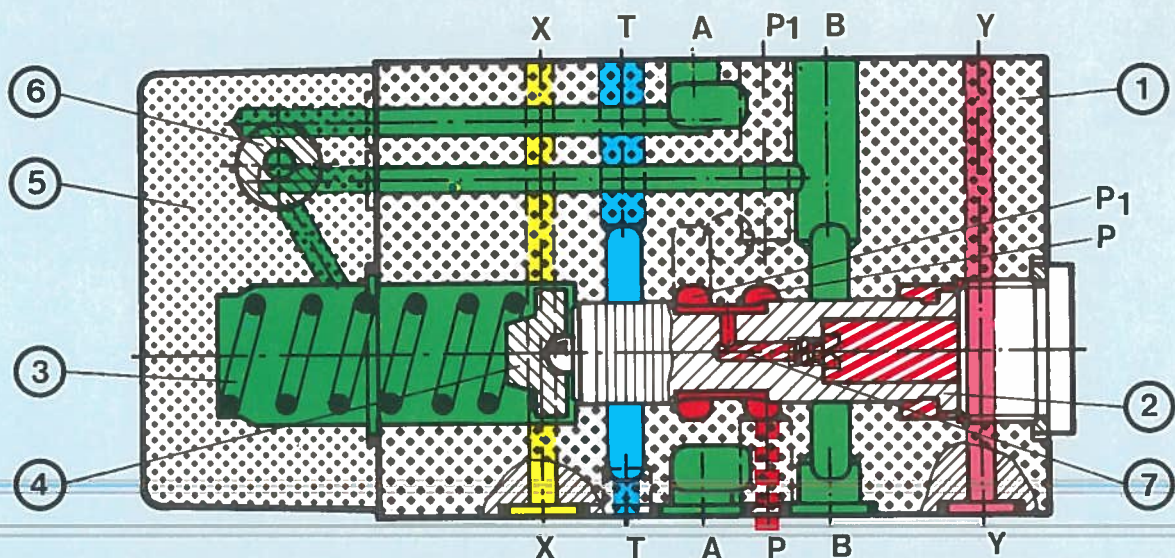


Fig. 4: 2-way meter-in pressure compensator in P-port, Type ZDC

When using standard proportional valves without pressure compensation, a volumetric flow resolution of 1 to 20 is still obtained in valves with spring feedback or 1 to 100 in valves with electrical feedback. This range can be considerably extended by the use of a pressure compensator. Fig. 5 shows curves which indicate the resolution of the volumetric flow of a typical proportional valve with pressure compensator. In the illustrated case, a volumetric flow resolution of 1 to 300 has been achieved, the pressure/flow characteristic is good over the entire range.

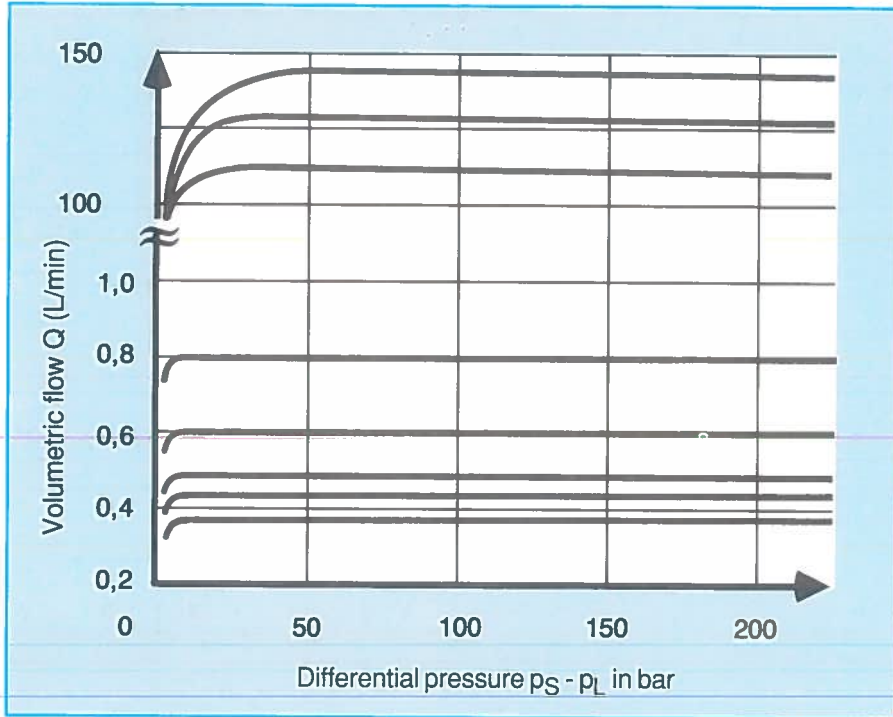


Fig. 5 Volumetric flow resolution of a proportional directional valve with meter-in pressure compensator

As the flow increases, the outer pressure difference ( $p_S - p_L$ ) must also increase in order to achieve the flow control function, i.e. the flow is no longer dependent on  $\Delta p$ .

Fig. 6 shows the dependency of this outer pressure difference on flow.

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If it were decided to operate with  $Q = 100$  L/min, and a load pressure of  $p = 120$  bar, a pump pressure of:

$$p_P = p_{\text{Load}} + p_{\text{min}} = 120 \text{ bar} + 22 \text{ bar} = 142 \text{ bar}$$

would be required.

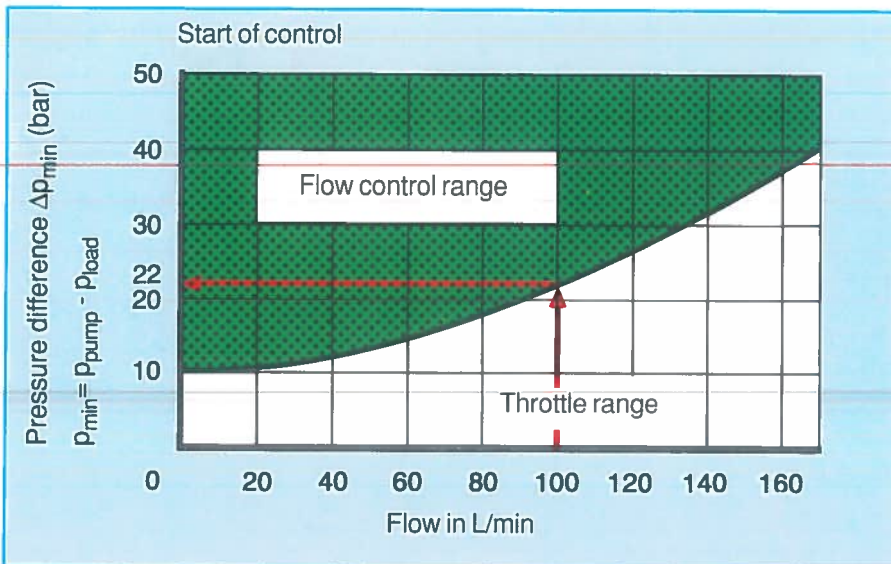


Fig. 6  $p_{\text{min}}$  / flow curve

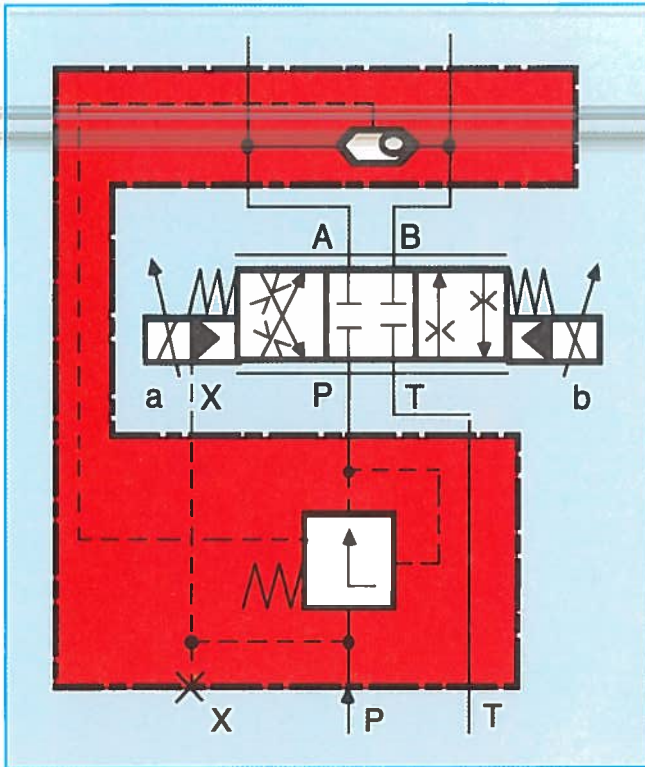


Fig. 7  
Pilot operated proportional directional valve 4WRZ with meter-in pressure compensator ZDC - internal pilot oil feed - sandwich plate design

When the sandwich plate pressure compensator is used in conjunction with pilot operated proportional directional valves, in principle the proportional valve with "external pilot oil supply" should be used. The "internal or external pilot oil supply" version of the pressure compensator can be used. With direct operated proportional directional valves, the pressure compensator with "external pilot oil" **must** be used. Oil must not reach port X as in the directly operated proportional directional valves, there is no seal at this point.

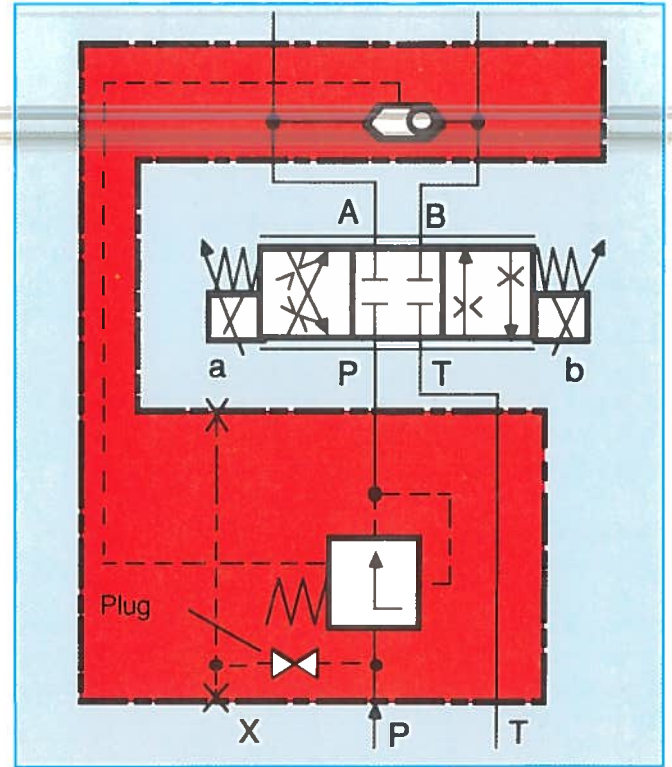


Fig. 8a  
Directly operated proportional directional valve 4 WRE with meter-in pressure compensator ZDC - external pilot oil feed - sandwich plate design

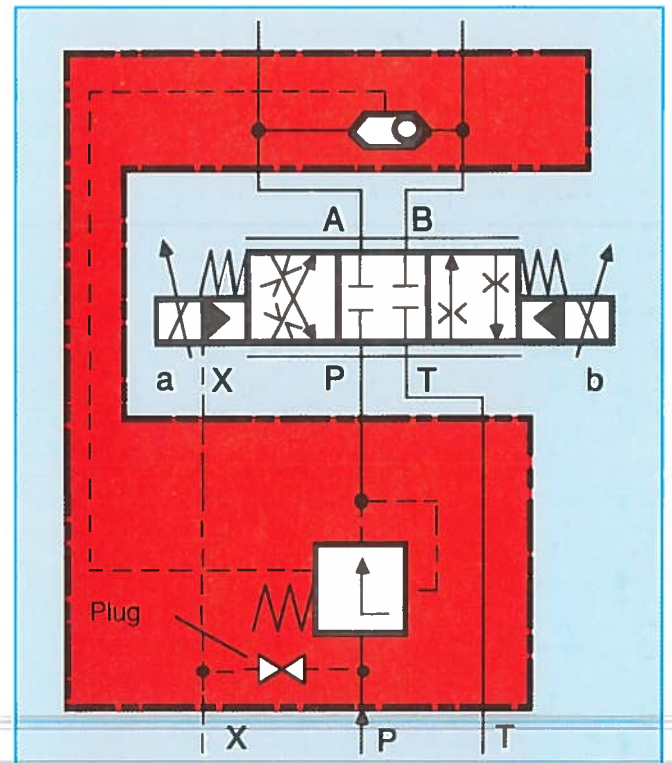


Fig. 8b  
Pilot operated proportional directional valve 4 WRZ with meter-in pressure compensator ZDC - external pilot oil feed - sandwich plate design

## Load Compensation with 3-way Meter-in Pressure Compensator

Up until now, 2-way meter-in pressure compensators have been discussed which are primarily used in industrial systems. 3-way meter-in pressure compensators (Fig. 10) are used more rarely despite their increased degree of efficiency. They can, however, in some cases be produced relatively easily by changing the spool in 2-way meter-in pressure compensators. The load application point corresponds to that of the 2-way meter-in pressure compensator. The resolution capacity and pressure/flow characteristic are identical to those of the 2-way meter-in pressure compensator. They are mainly used in conjunction with fixed pumps.

### Function of 3-way Meter-in Pressure Compensator

When using the 3-way meter-in pressure compensator, the fixed measuring orifice A2 and the orifice opening A1 controlled by the pressure compensator are arranged in parallel.

The control orifice A1 releases an outlet opening.

The following applies with regard to the balanced position of the control spool: Without taking into consideration the frictional and flow forces.

$$p_1 \cdot A_K = p_2 \cdot A_K + F_F$$

$$\Delta p = \frac{p_1 - p_2}{F_F / A_K} = \text{const.}$$

The pressure drop is once again held constant at the measuring orifice, thereby achieving a flow Q independent of the changes in pressure.

Contrary to the 2-way pressure compensator where the pump must constantly produce maximum pressure, when using the 3-way pressure compensator, the working pressure is greater than the driven unit pressure only by the amount of the pressure drop  $\Delta p$  at the measuring orifice.

As a result, the power loss is considerably less. If a W- spool is used in the proportional valve (A and B in centre position linked to tank) a bypass is provided from the pump to the tank with a retaining force amounting to the differential control pressure.

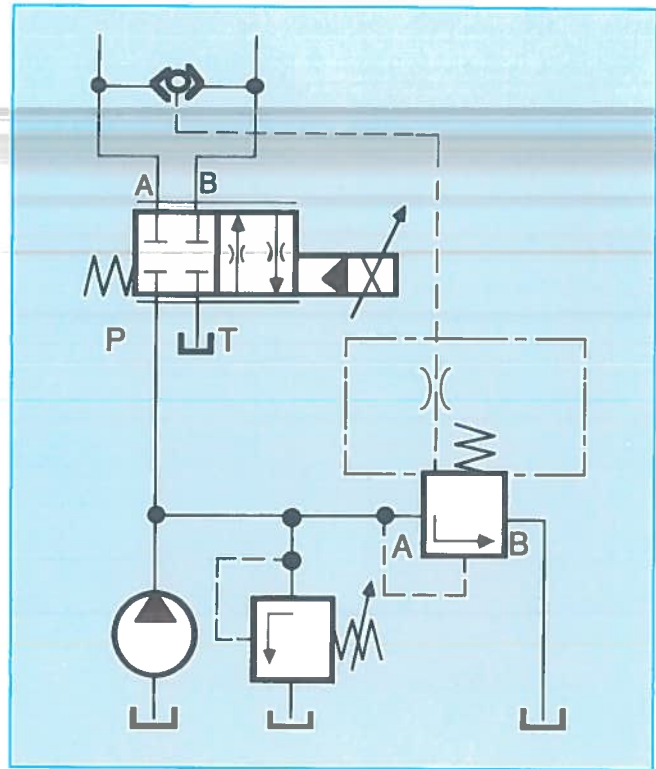


Fig. 9 Circuit example

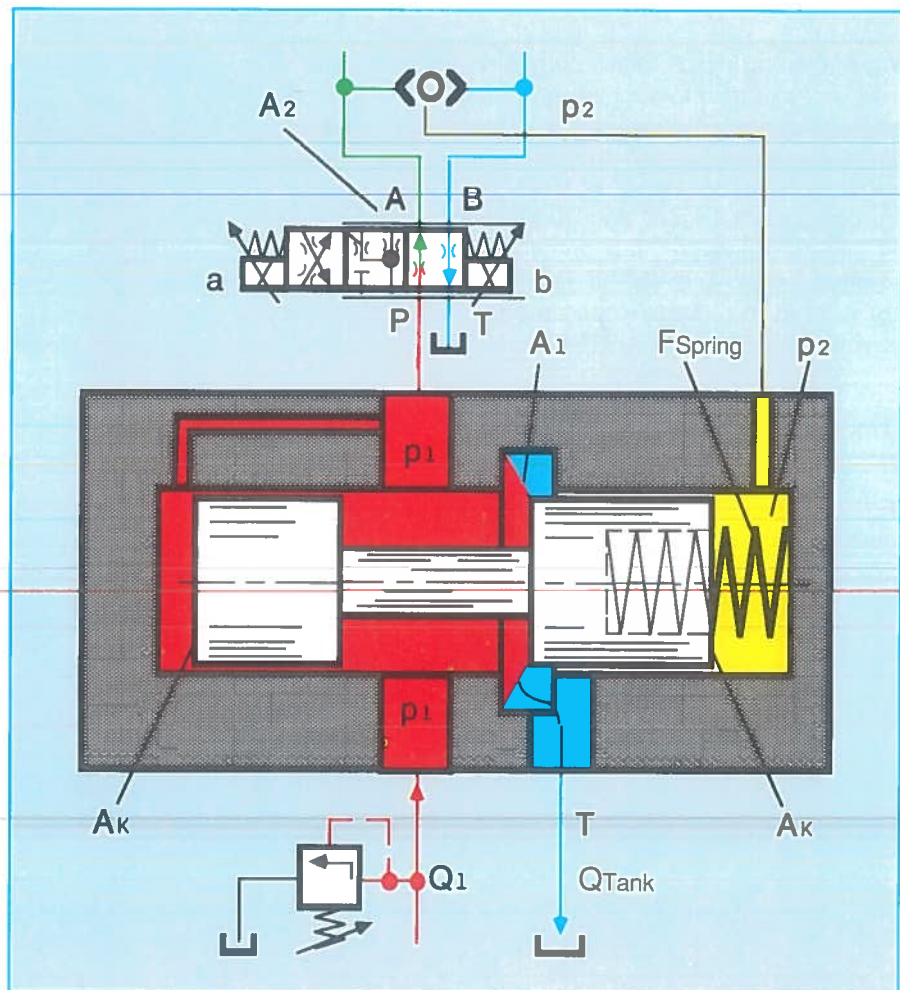


Fig. 10 Principle diagram

### Important Information on the Use of the Meter-in Pressure Compensator

Meter-in pressure compensators are known to have the disadvantage of not operating correctly under conditions involving deceleration, or counter balancing. Particularly when the de-celeration pressures are higher than the pressure drop defined by the spring for the meter-in throttle edge.

Circuits equipped with a shuttle valve no longer signal the pressure on the inlet side (A) during the deceleration phase but rather the pressure on the outlet side (B) (Fig. 11) which at this moment is higher, thereby causing the pressure compensator to open. As a result, the volumetric flow through the proportional valve increases.

The drive has a tendency to accelerate, however, the closing movement of the proportional valve acts against this. Cavitation is effectively prevented on the feed side.

The drive is decelerated by a simple throttling effect (not flow controlled).

In circuits without a shuttle valve, cavitation can occur due to holding the inlet pressure drop constant. This can cause considerable damage, particularly when a retaining force such as a deceleration valve (Fig. 13) or pressure control valve (Fig. 12).

If none of the two retaining forces is provided, the use of the meter-in pressure compensator must be restricted to drives with distinctly positive direction of load.

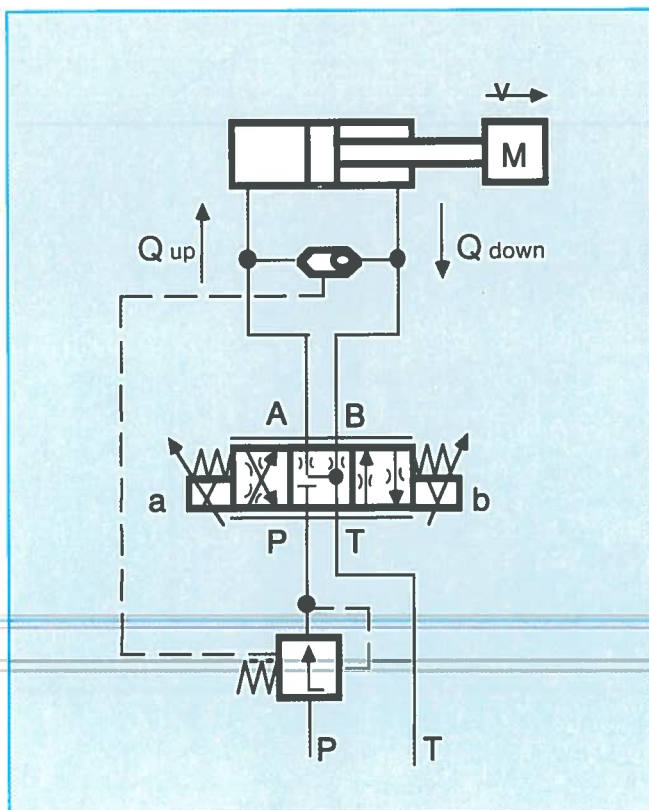


Fig. 11 Pressure control valve as retaining force

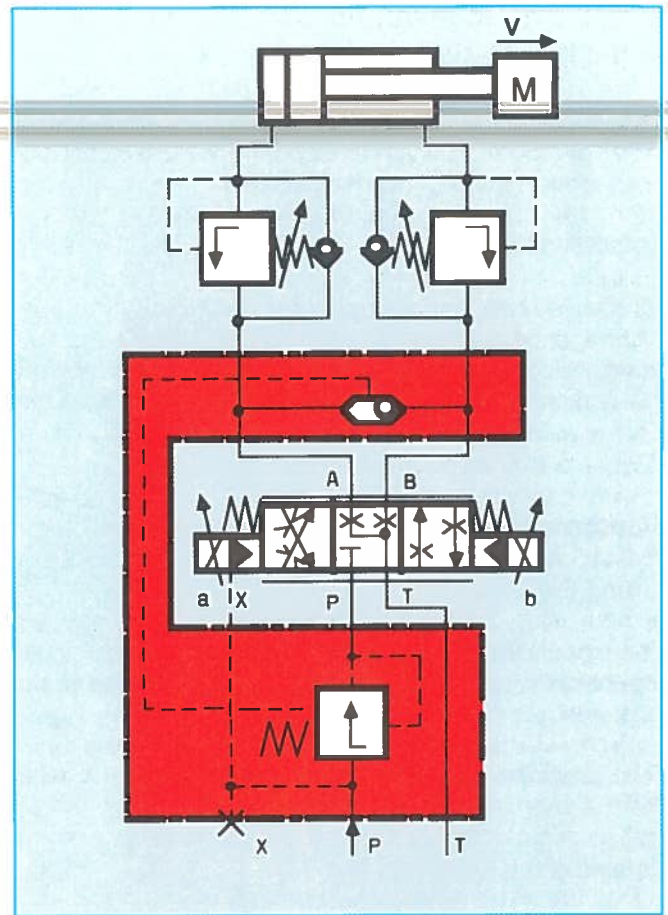


Fig. 12 Meter-in pressure compensation

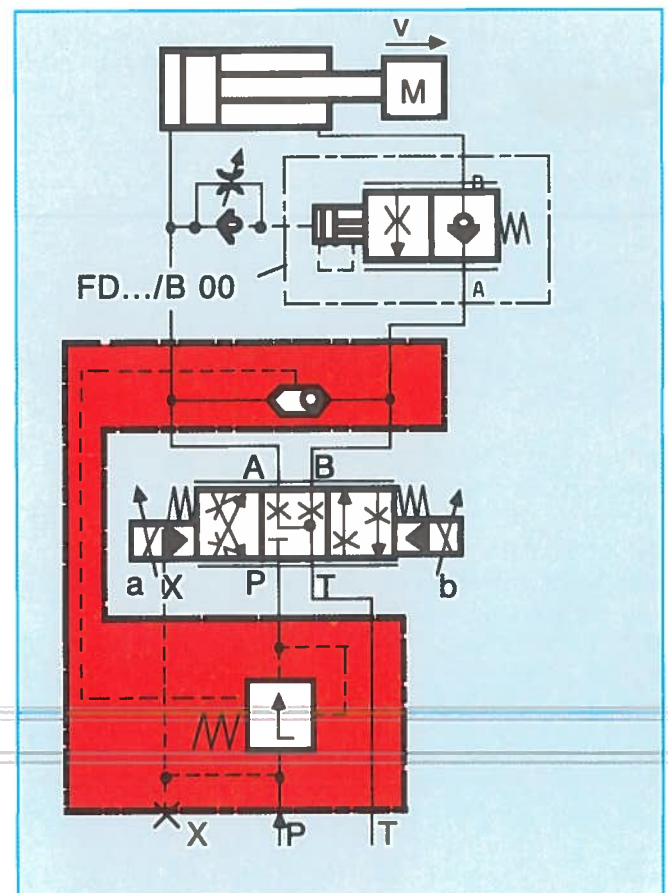


Fig. 13 Deceleration valve as retaining force

### Deceleration Valve Type F (Check-Q-Meter)

The check-Q-meter basically consists of the housing (1), main spool (2), auxiliary spool (3), control spool (4), drag spool (5) and control damper (6).

Its functions are:

- Controlled check valve, free of leakage oil
- Q-meter; it controls the outgoing oil flow  $Q_2$  corresponding to the oil flow  $Q_1$  fed on the opposite side of the consumer. The area ratio must be observed in the case of cylinders ( $Q_2 = Q_1 \cdot \phi$ ).
- Bypass valve due to free flow in opposite direction
- Secondary pressure relieve valve by means of additional attachment (only flange version possible).

#### Lifting the Load

The main spool (2) is opened for free flow from A to B. The main spool (2) is immediately closed if the pressure drops below the load pressure (e.g. pipe break between directional control valve and port A). This function is obtained by connecting the load side (7) with the chamber (8).

#### Lowering the Load (See Fig. 14)

The direction of flow is from B to A. Port A of the check-Q-meter is linked to the tank via the directional control valve. A quantity of oil is applied to the piston side of the cylinder corresponding to operating conditions.

The ratio pilot pressure at port X : Load pressure at port B = 1:20.

When the pilot pressure is reached at port X (1/20 of load pressure) the main poppet is preopened; the ball in the main poppet is raised from its seat by the control spool (4).

As a result, the chamber (8) is depressurized via the hole in the auxiliary piston (3) and via side A to the tank. At the same time, the load pressure applied to the chamber (8) from chamber B is interrupted by the longitudinal movement of the auxiliary spool (3) in the main poppet. The pressure at the main poppet (2) is relieved. During this procedure, the position of the control spool (4) is such that its face end is resting on the main poppet (2) and its collar on the drag spool (5). The pressure at port X necessary for opening B to A is now only influenced by the spring in chamber (9). The initial pressure for opening the connection B to A is 20 bar; a pressure of 50 bar is required for complete opening.

The relationship between pilot pressure, opening area and the differential pressure over the connection B to A determines the outlet oil directly dependent on the inlet oil at a driven unit, so that uncontrolled advance of the driven unit is not possible.

It is normally possible to influence the opening and closing characteristics of the deceleration valve by the use of a throttle/non return valve in the X line - meter-out throttling.

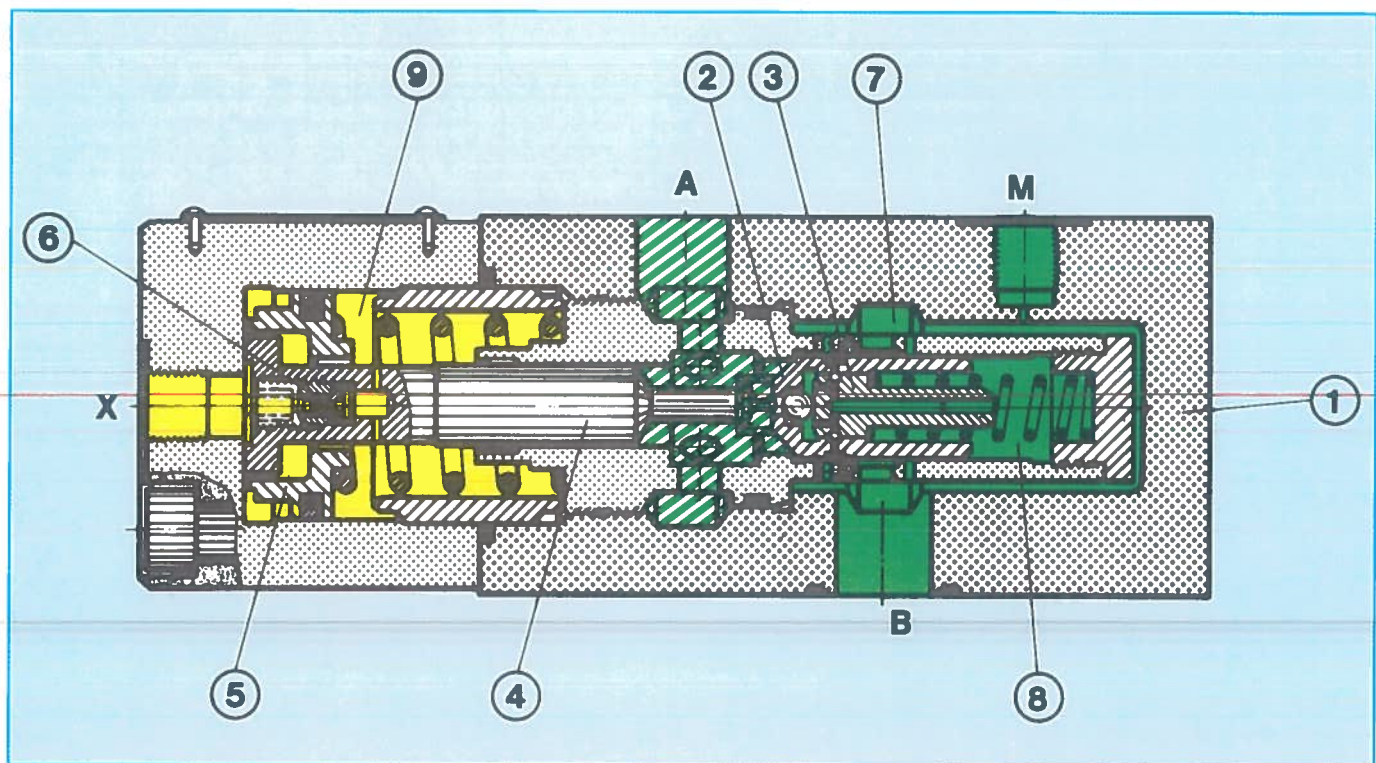


Fig. 14 Check-Q-Meter

## System Supplements

### 1. Max. Pressure Limitation

Max. pressure limitation for the drive can be achieved if the spring chamber of the compensator is connected to a pressure relief valve as shown in Fig. 15.

### 2. Variable $\Delta p$

As already described, the pressure drop across the throttle is initially determined by the pre-load of the built-in spring.

The differential pressure can be infinitely varied at the throttle edge if the load application point is directed as shown in Fig. 16 via a pressure relief valve.

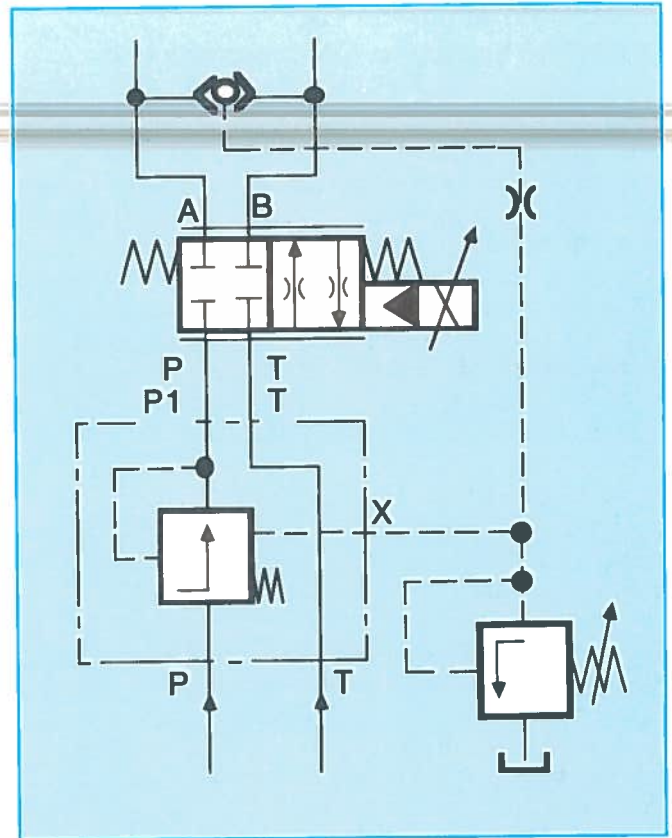


Fig. 15 Meter-in pressure compensator with max. pressure limitation

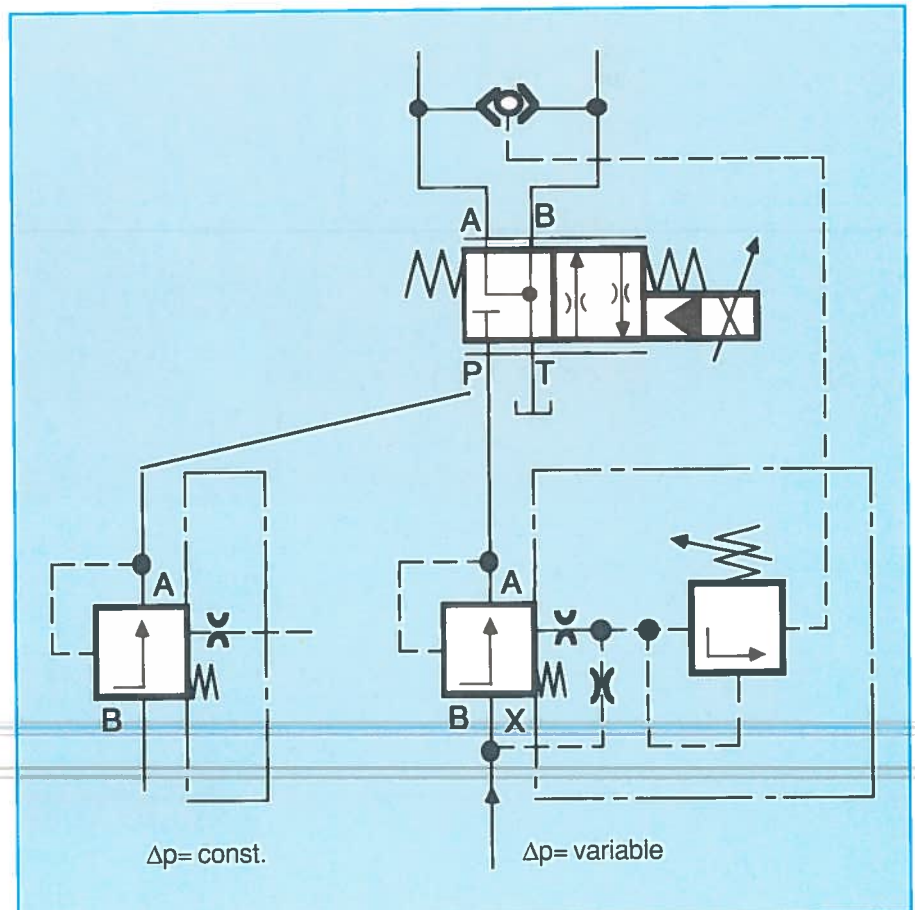


Fig. 16  $\Delta p$  variable via pressure relief valve

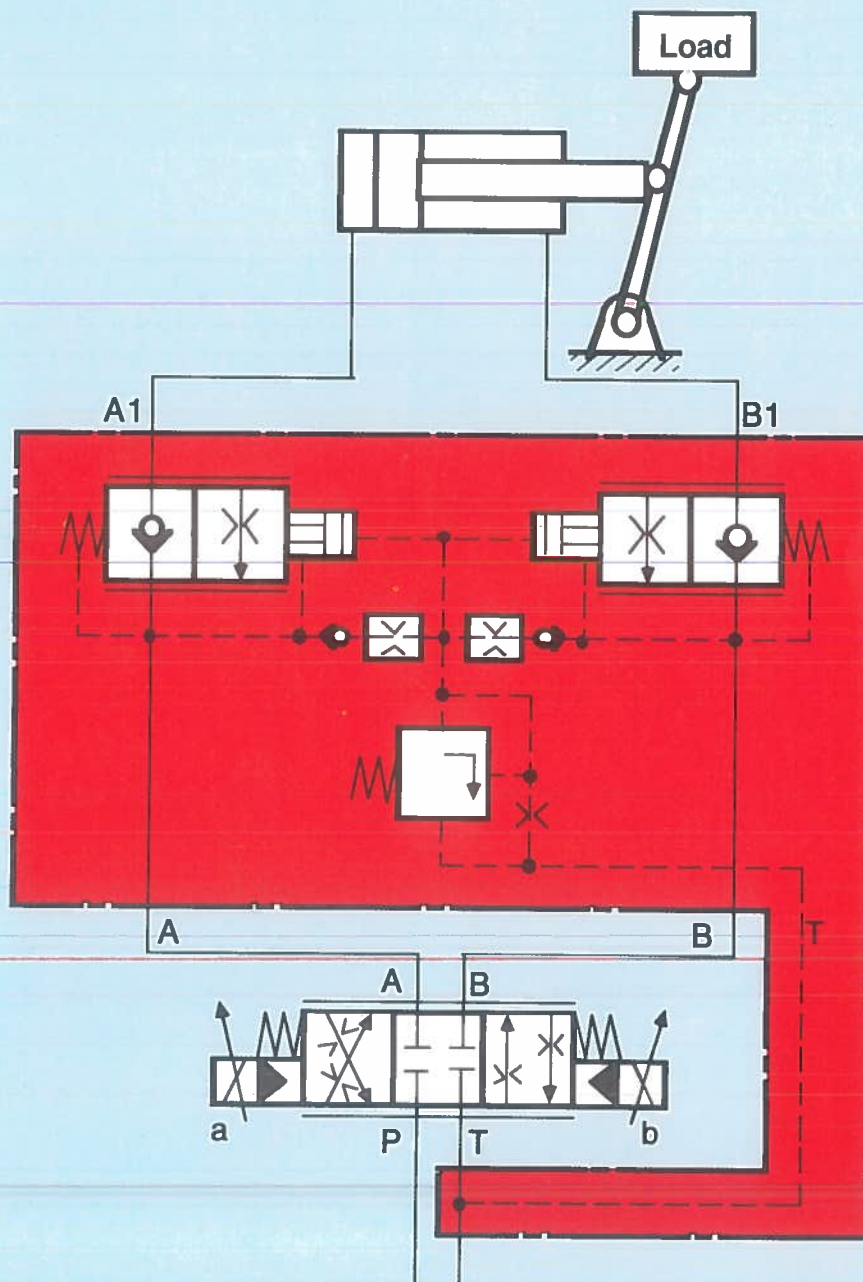
## Load Compensation with Meter-out Pressure Compensators

In systems in which the direction of applied load reverses, the use of meter in compensators is severely restricted. In such cases, a meter-out pressure compensator is often used, arranged in one or in both connections of the driven unit depending on the application.

The meter-out pressure compensator is always mounted in the outlet between the driven unit and the proportional valve and maintains the pressure drop constant from A or B to the tank.

Meter-out pressure compensators of poppet design are available for the sizes 16, 25 and 32 instead of the commonly used spool valves. This arrangement therefore combines the compensator function and the function of the pilot operated check valve normally required to support vertical loads, since these pressure compensators leak-free. The poppets simply lift to allow flow in the opposite direction thereby making by-pass checks unnecessary.

Fig. 17 Circuit example meter-out pressure compensator of poppet valve design



## Meter-out Pressure Compensator with Cut-off Function

The unit basically consists of the housing (1), the valve elements (2.1) and (2.2) as well as the pressure relieve valve (3).

The amount and direction of the oil flow are preset at the signal value potentiometer of the proportional directional valve.

If for example, the pump is switched to port A, the fluid flows via the valve element (2.1) to the driven unit. In this case, the valve element (2.1) functions as a non-return valve. At the same time, the flow of pilot oil is derived from the flow delivered by the pump and routed to the chamber (5) via the control spool (4.1) acting as a load compensating flow control valve. This flow of pilot oil builds up a pressure in front of the pressure relief valve (3) which is applied to the B side of the control spool (4.2) via the orifices (6) and (7).

In addition, the outlet of the pressure relief valve is linked to the channel T. The control spool (4.2) opens

the pressure relief poppet (8) against the load pressure applied in the spring chamber (9) (max. 315 bar). At the same time, the pressure relief poppet (8) closes off the link to the load pressure. In the spring chamber (9) the pressure is applied via the pressure take-off at the relief poppet (8) in front of the proportional directional valve in channel B. This pressure also acts on the annulus side and the face area of the control spool (4.2).

The pressure drop from B to T over the proportional directional valve is therefore constant. This pressure drop is controlled by the control land (10) and represents the pressure difference in chamber (11) minus the spring force (12). The force of the spring (13) is of no significance.

The valve element (2.1) in A functions as previously described when the proportional directional valve switches the pump to B.

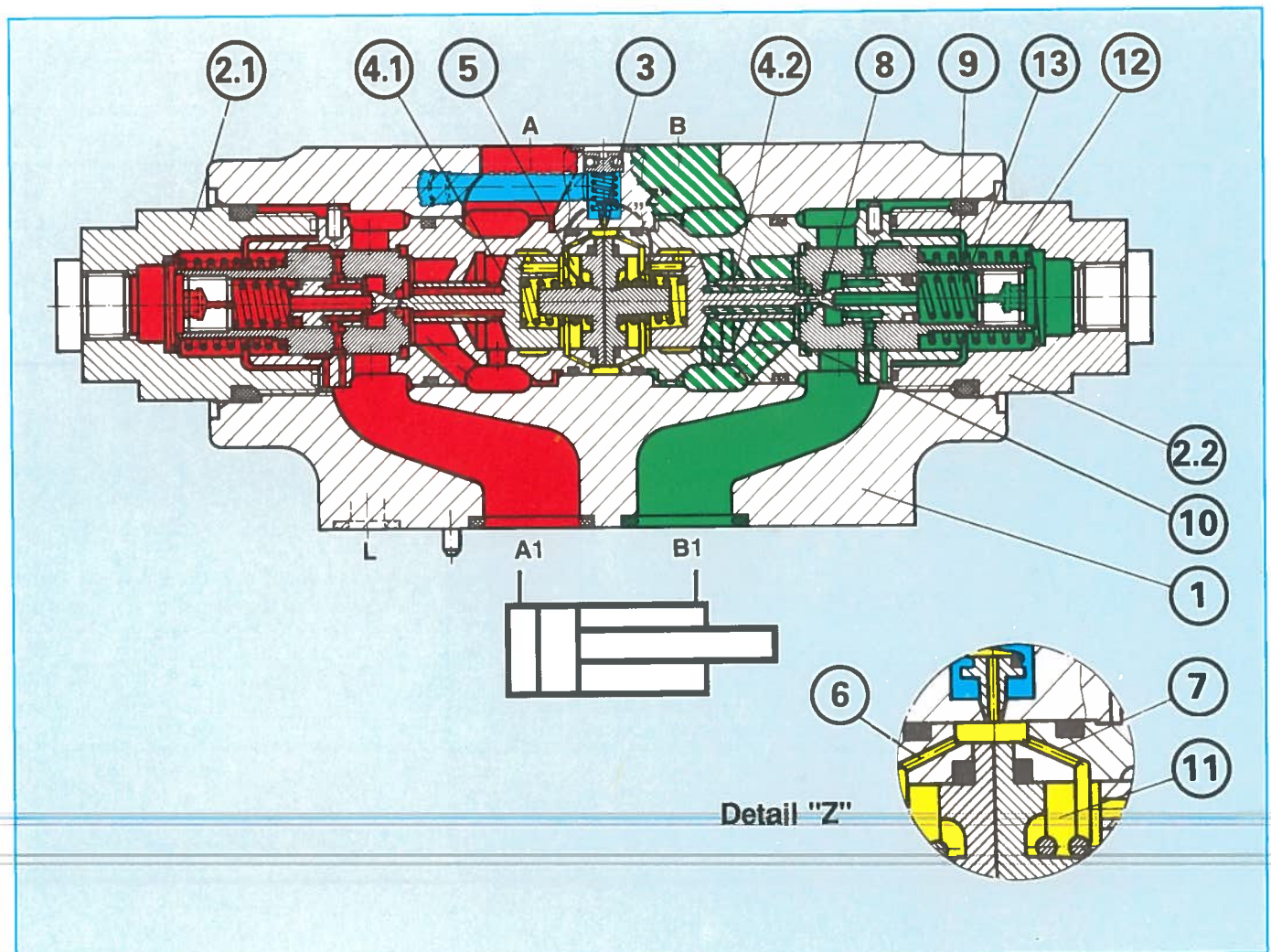


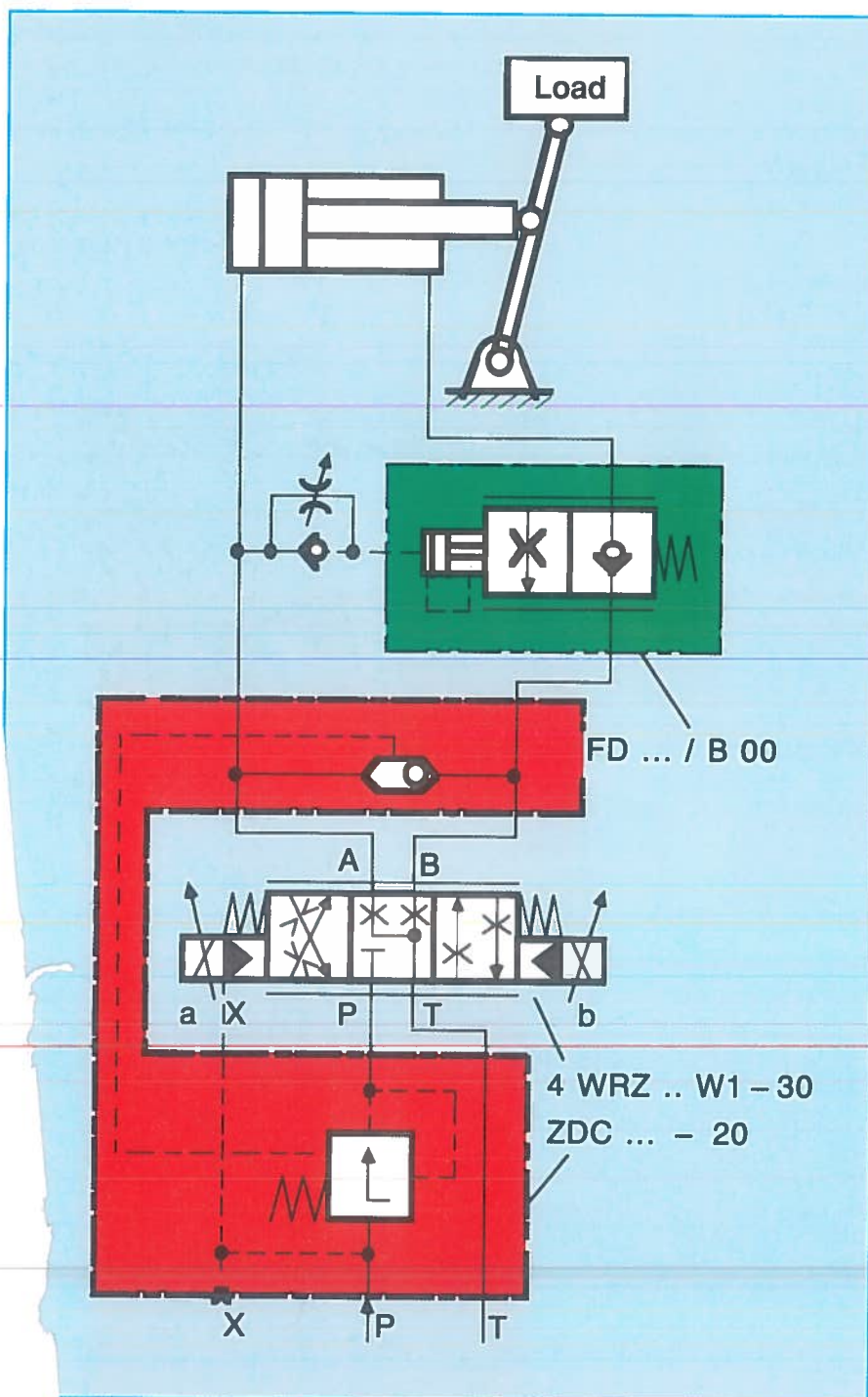
Fig. 18 Sectional view of meter-out throttle isolating pressure compensator of poppet design

### Caution

When using the meter-out compensators with cylinders with differing area ratio, there is the danger of pressure intensification (see flow control valve in outlet) to the rod side of the cylinder.

If pressure intensification is likely to occur, a meter in pressure compensator should be used with a suitable load compensating valve.

load compensating valve.



9 Meter-in pressure compensator with brake valve

## Application Limits and Circuit Options

### Which open loop controls can be realized with the meter-out pressure compensator?

All open loop controls for hydraulic motors, cylinders with double piston rod or cylinders with single-sided piston rod provided the pressure intensification on the cylinder annulus side defined by the meter-out throttle isolating pressure compensator is acceptable.

### Which open loop controls are not possible with the meter-out pressure compensator?

A meter-in pressure compensator must be provided if pressure intensification on the annulus side is to be avoided. The check-Q-meter on the B side acts as a load holding valve (see Fig. 19).

The regenerative circuit (Fig. 20) cannot be implemented with the meter-out pressure compensator. A meter-in pressure compensator is necessary for this purpose.

While the cylinder is extending, the maximum deceleration pressure corresponds to the pump pressure and is normally sufficient.

In the control of a plunger cylinder (Fig. 21), a meter-in pressure compensator (red area) is necessary for the up stroke and a meter-out pressure compensator (green area) for the down stroke.

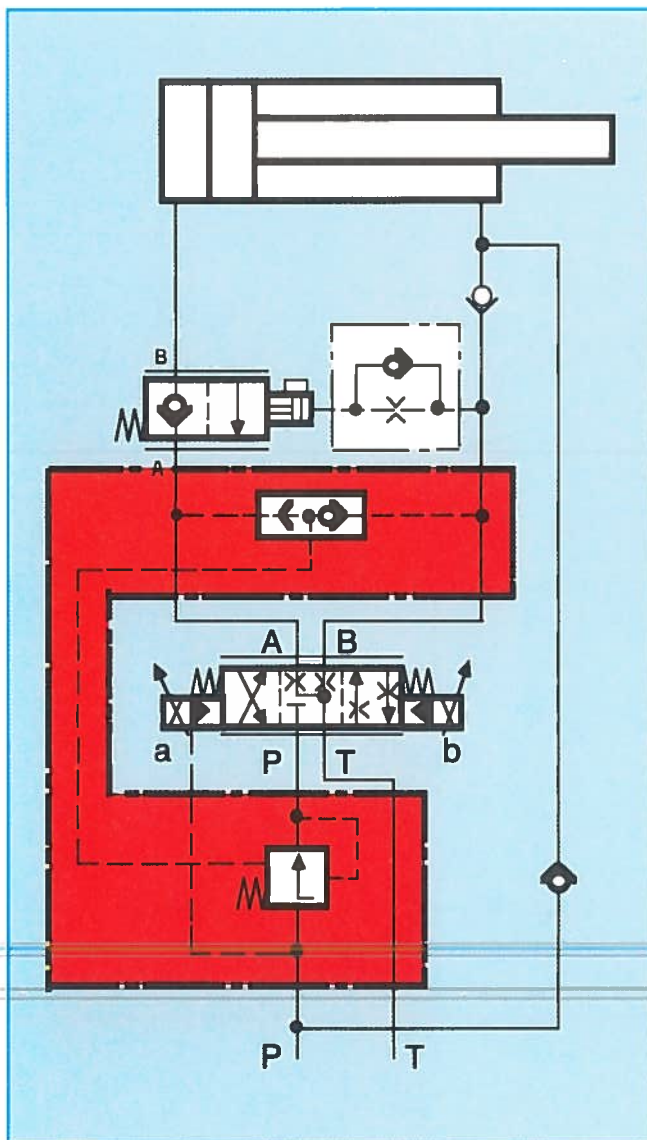


Fig. 20

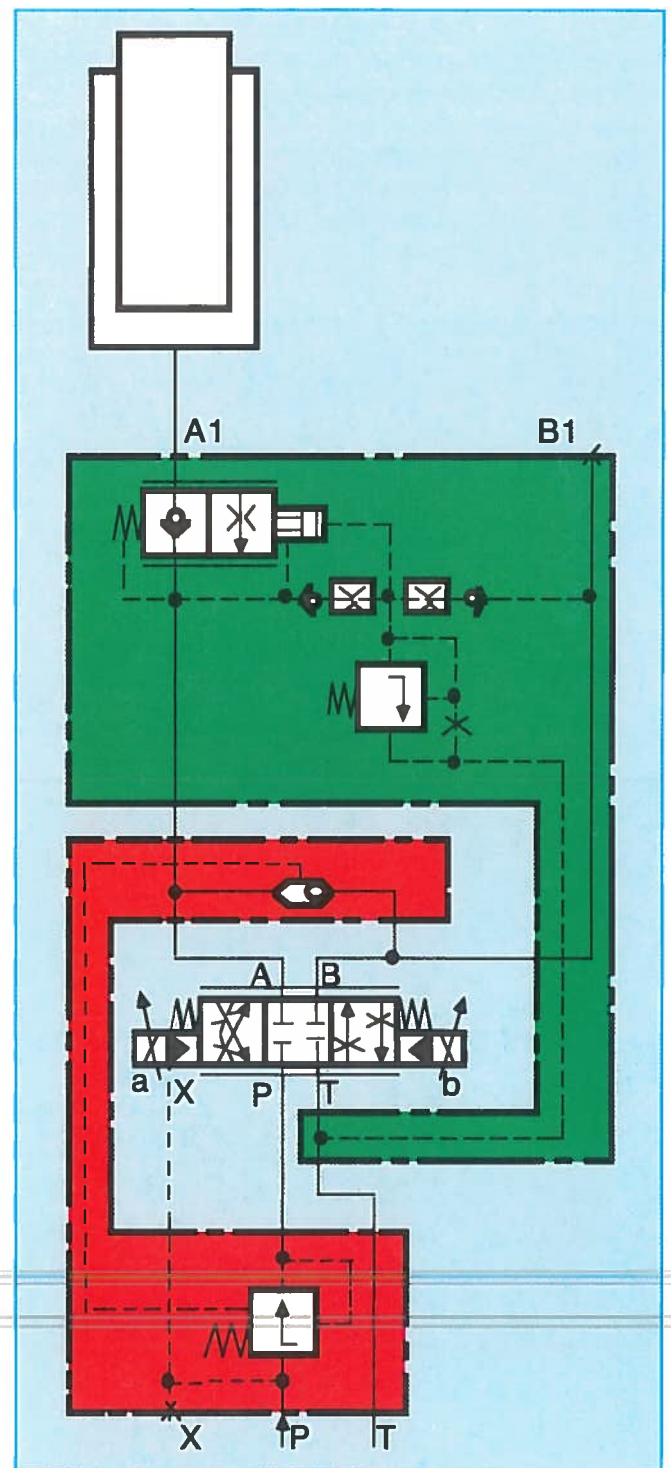


Fig. 21

Fig. 21

In the case of high flow rates, load compensation can be realized by means of 2-way cartridge valves (logic elements) with pressure reducing function (DR) or pressure limiting function (DB).

### 2-Way Pressure Compensator Pressure Reducing Logic Element

The 2-way valve with pressure reducing function must always be arranged in the direction of flow in front of the throttling point in order to obtain a constant pressure drop at the throttle.

The control lands of the 2-way valves have been modified for load compensation applications.

To ensure adequate damping properties of the 2-way valve, an orifice is normally installed in the cover. Orifice size is chosen to suit the valve size.

In various applications it is of advantage when the 2-way valve opens without damping and closes controlled via a jet. For this reason, a cover is available with

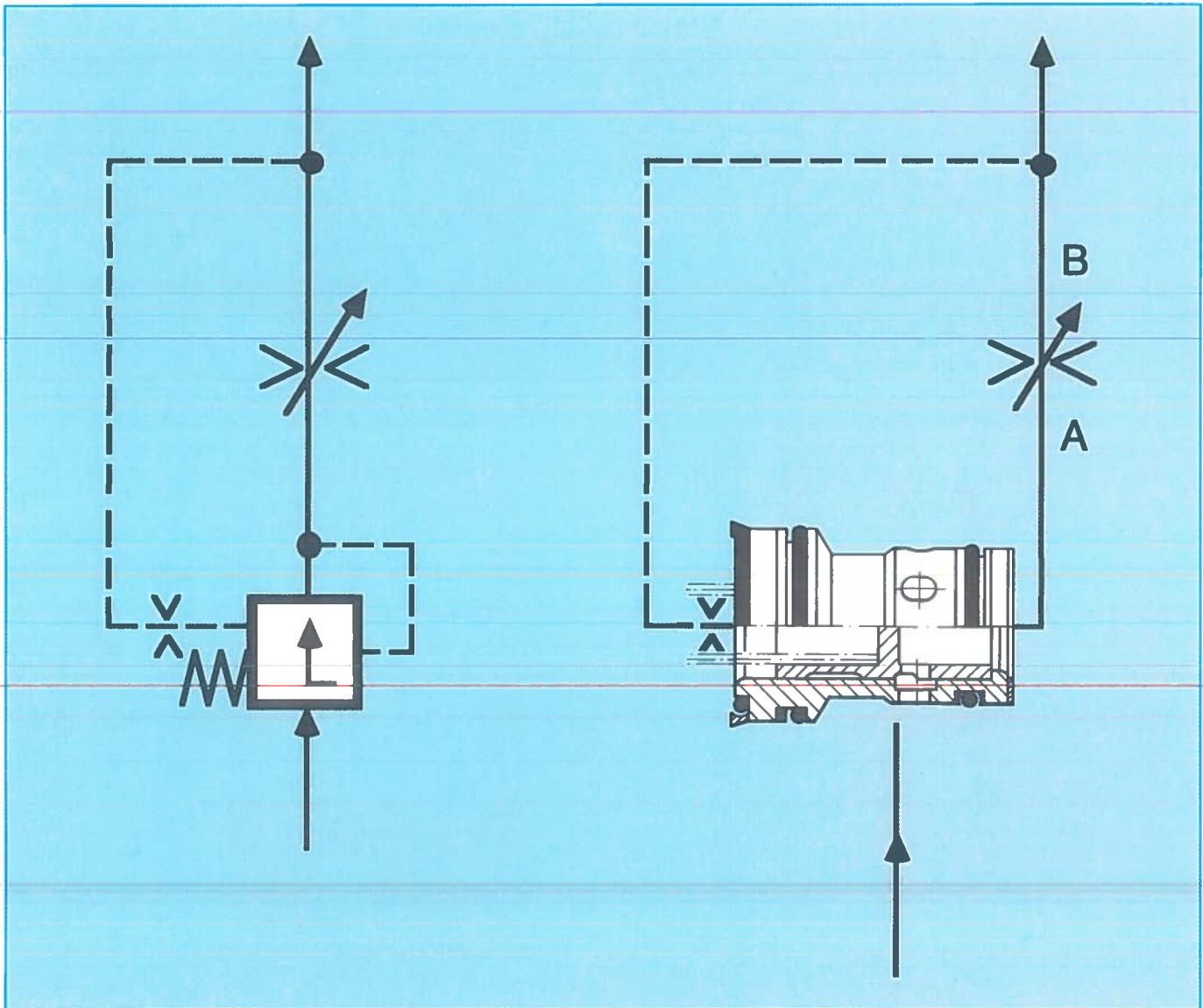


Fig. 22: 2-way valve for load compensation

## Guidelines in project engineering

### Circuit examples

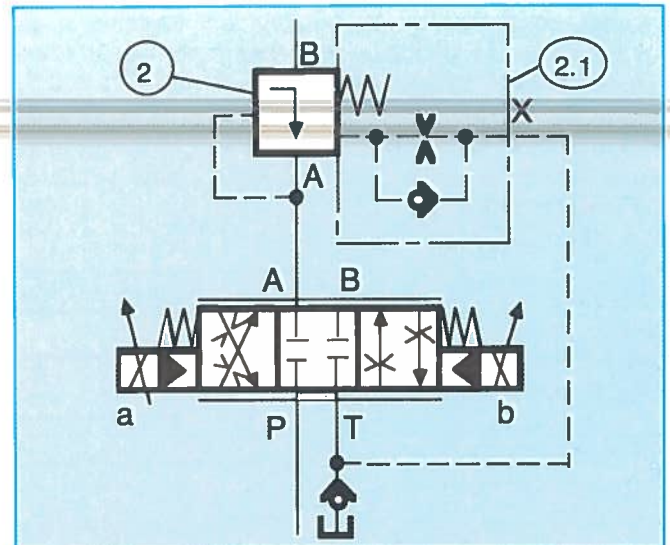


Fig. 24: 2-way meter-out pressure compensator  $\Delta p = 8$  bar

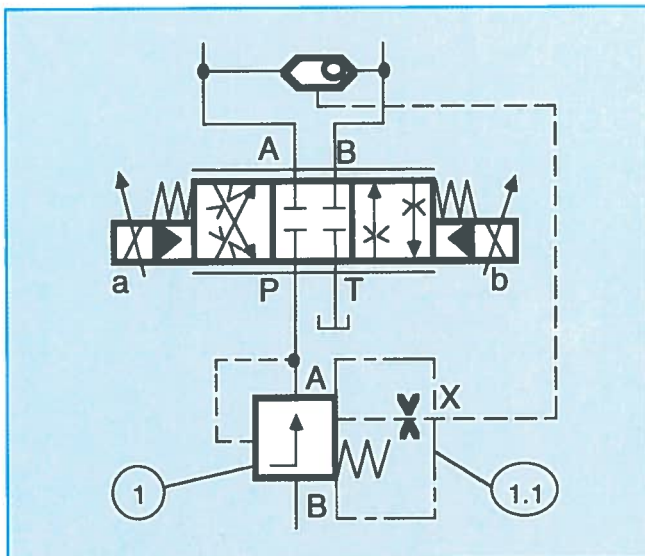


Fig. 23: 2-way meter-in pressure compensator  $\Delta p = 8$  bar

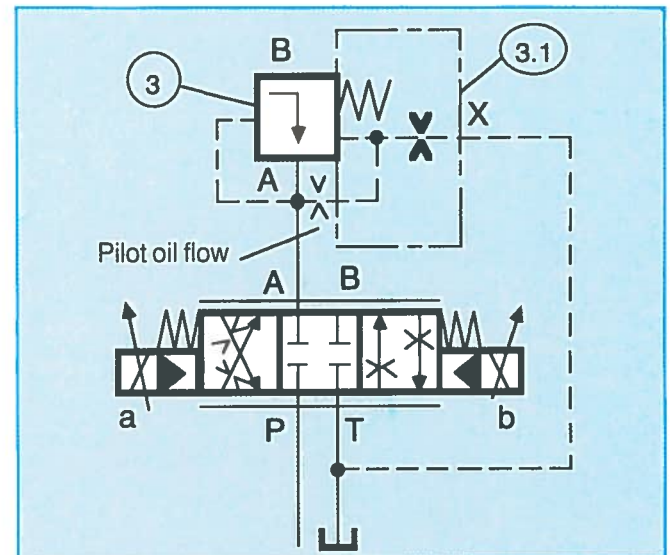


Fig. 25: 2-way meter-out pressure compensator  $\Delta p \approx 15 \dots 18$  bar

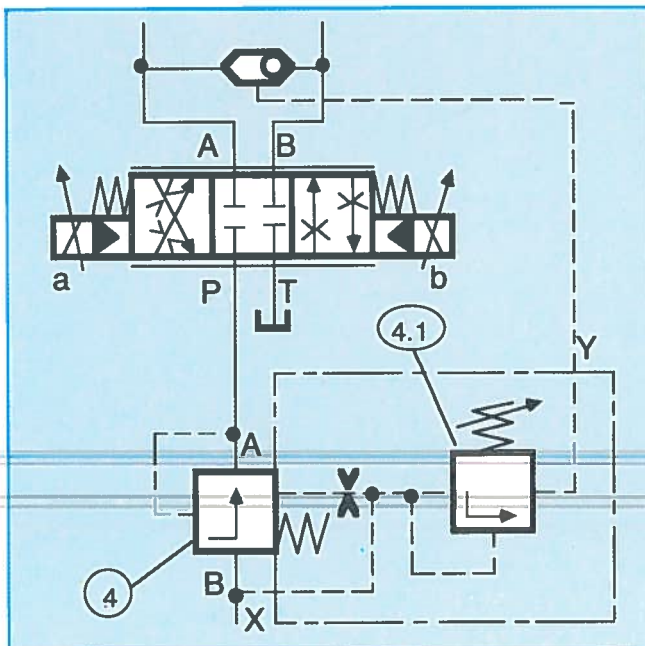


Fig. 26: 2-way meter-in pressure compensator  $\Delta p$  variable

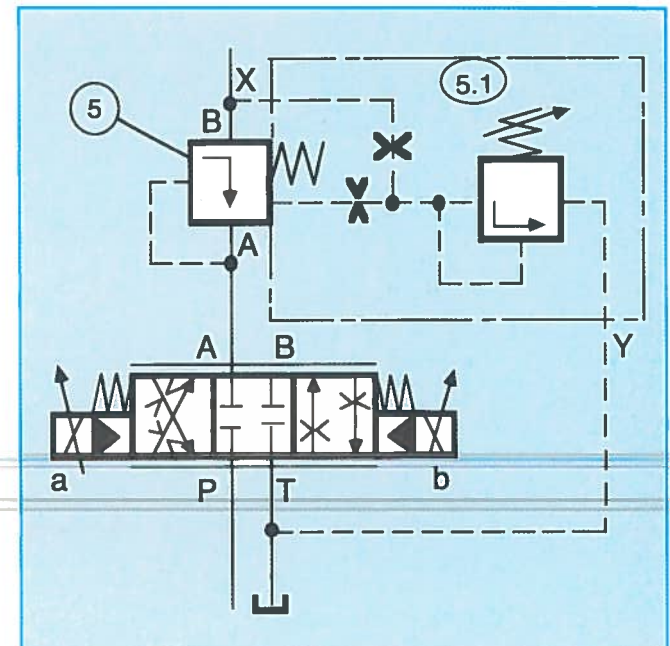


Fig. 27: 2-way meter-out pressure compensator  $\Delta p$  variable

## Load Compensation with Pressure Compensators

1) Load compensation for positive and negative loads for cylinders and hydraulic motors with no regenerative circuit, using logic elements.

Care must be taken in the case of cylinders with an area ratio=2:1 to ensure that the main spool of the proportional directional valve has the throttle opening ratio of 2:1.

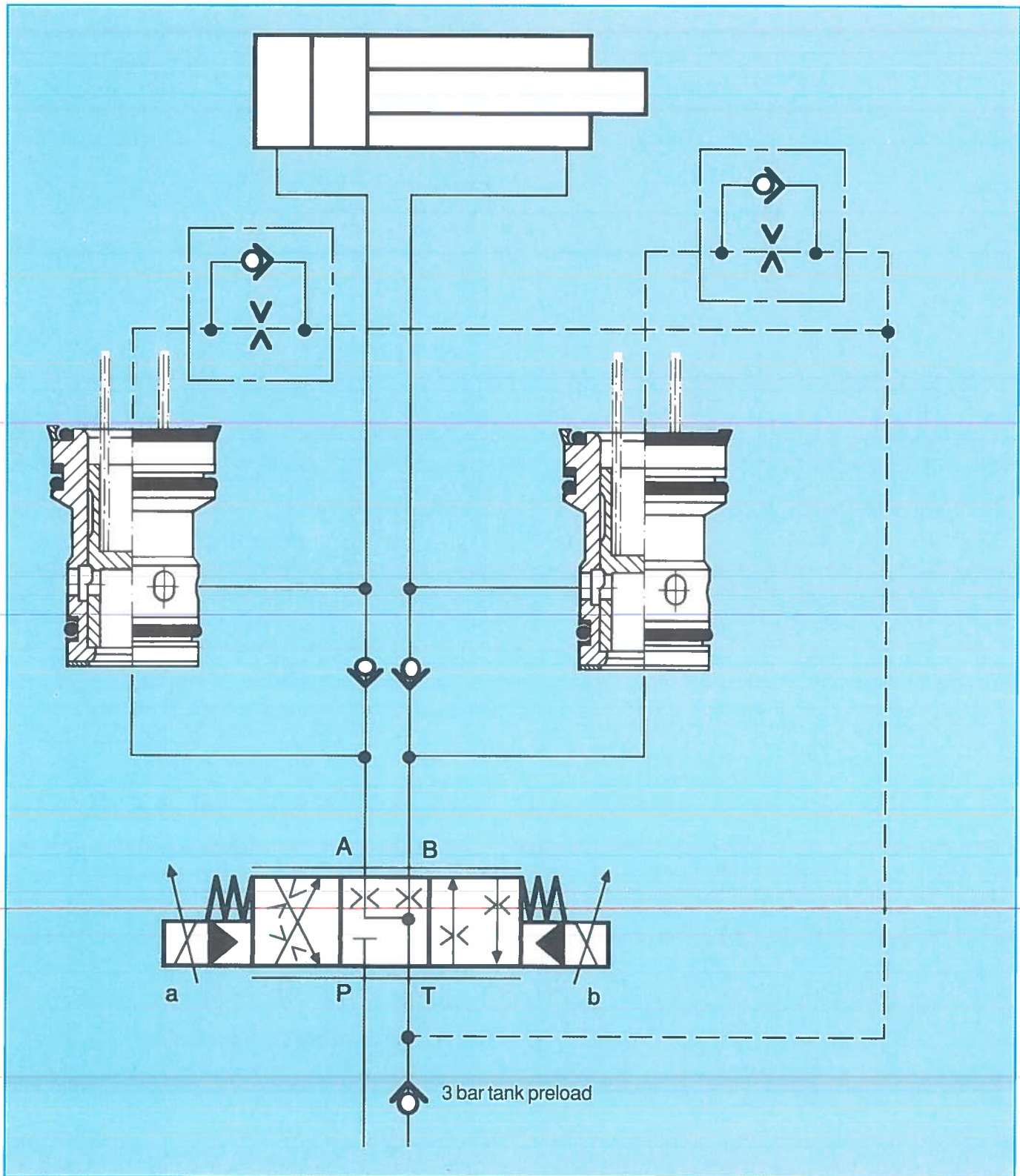


Fig. 28

2) Load compensation for positive and negative loads for cylinders with an area ratio of 2:1 with a generative circuit including logic elements.

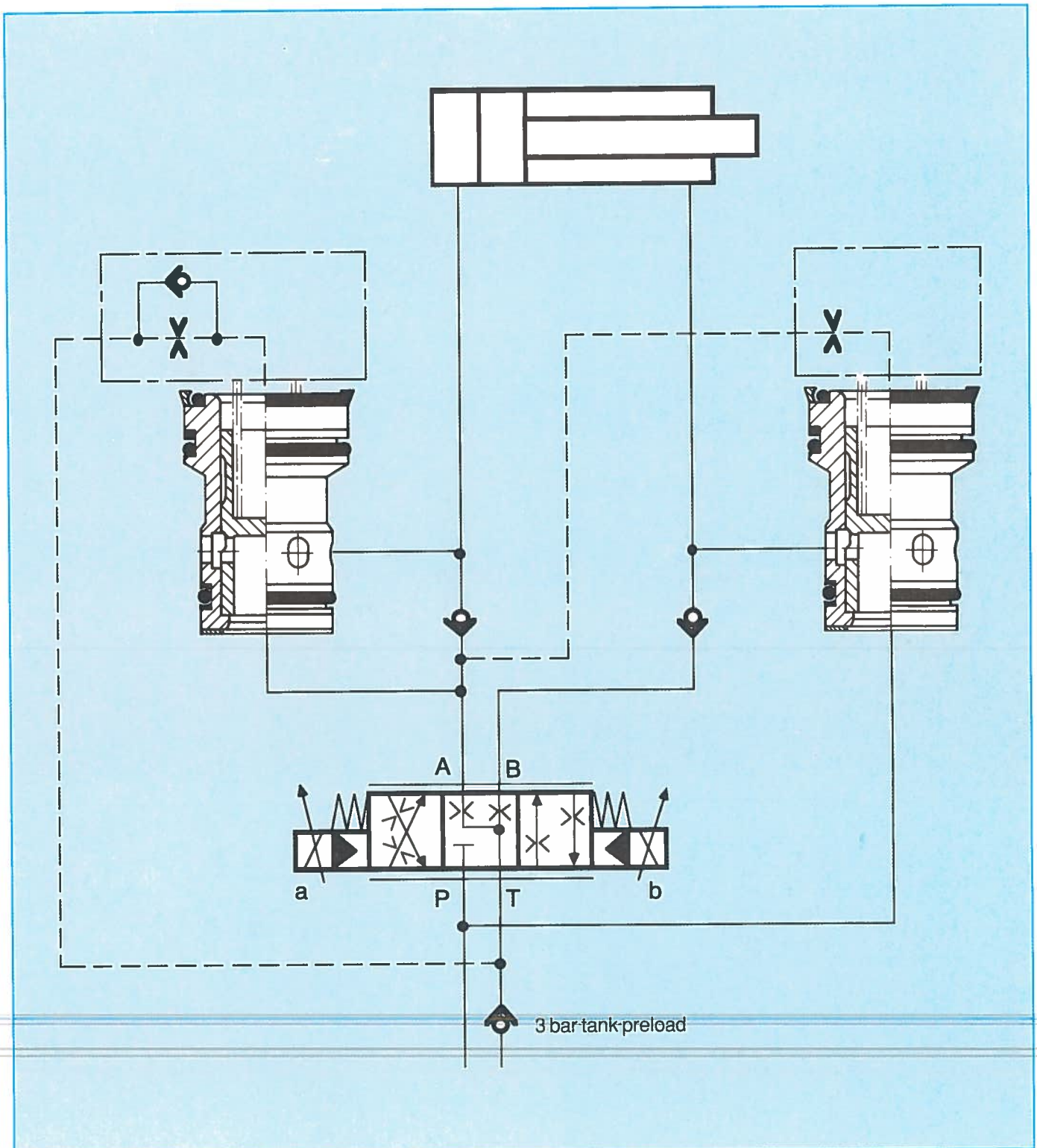


Fig. 29

### 3-Way Pressure Compensator in Pressure Limit Function

The built in valve for the pressure limiting function is designed as a spool/poppet valve without area difference (no effective area at port B). It is always arranged parallel to the throttling point.

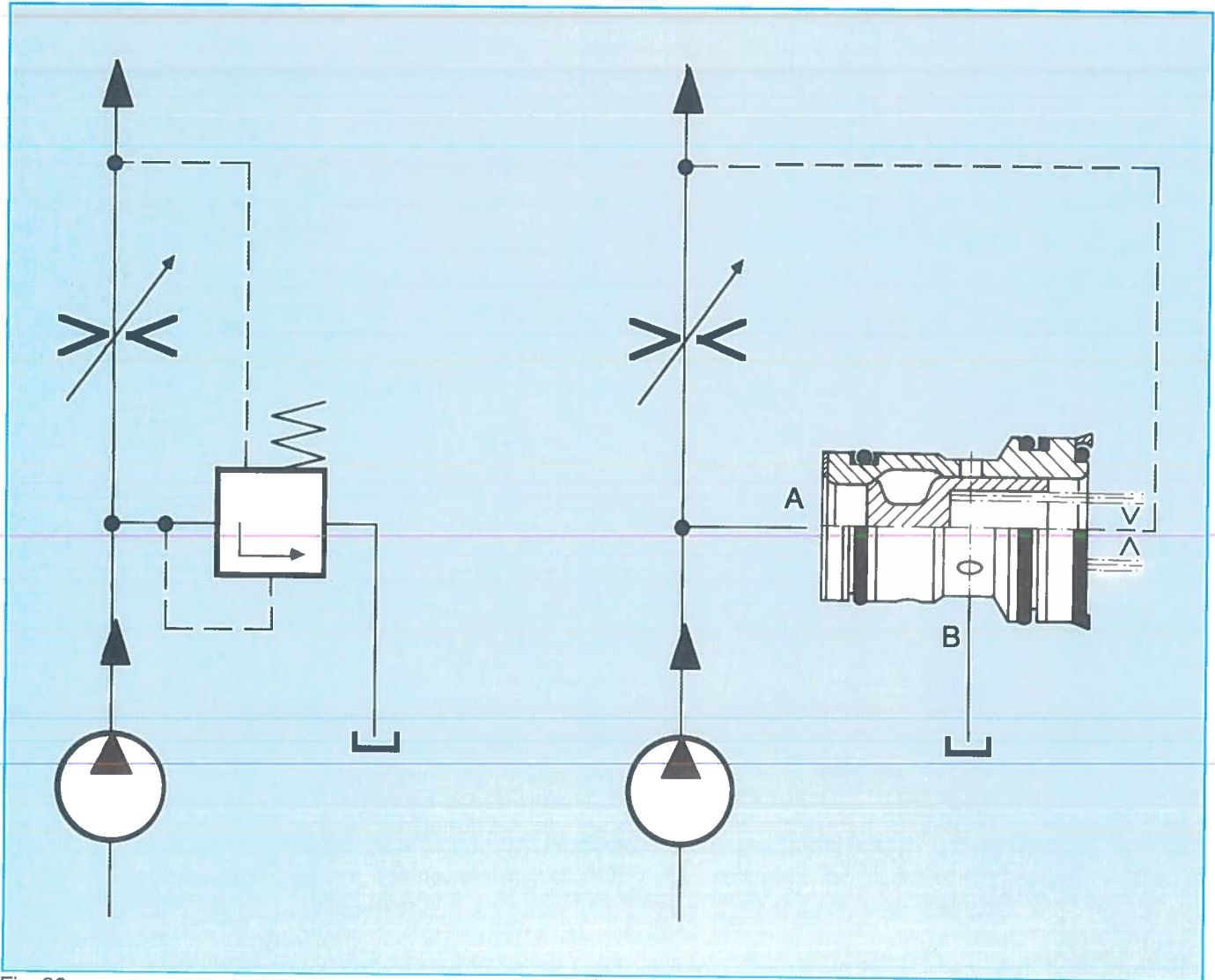


Fig. 30

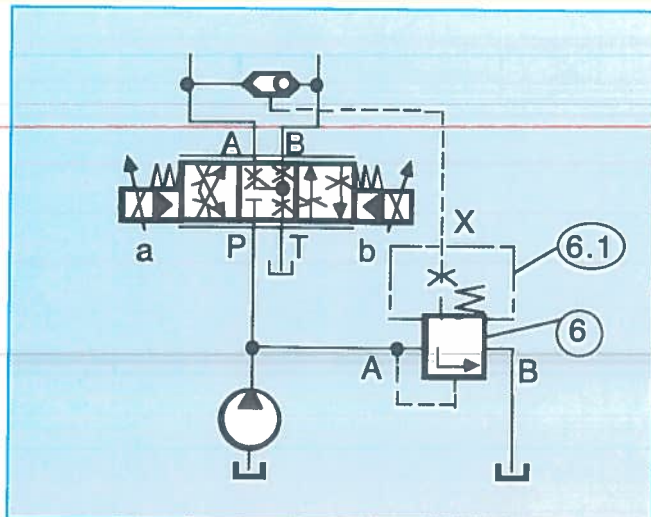


Fig. 31: 3-way pressure compensator  $\Delta p = 8 \text{ bar}$

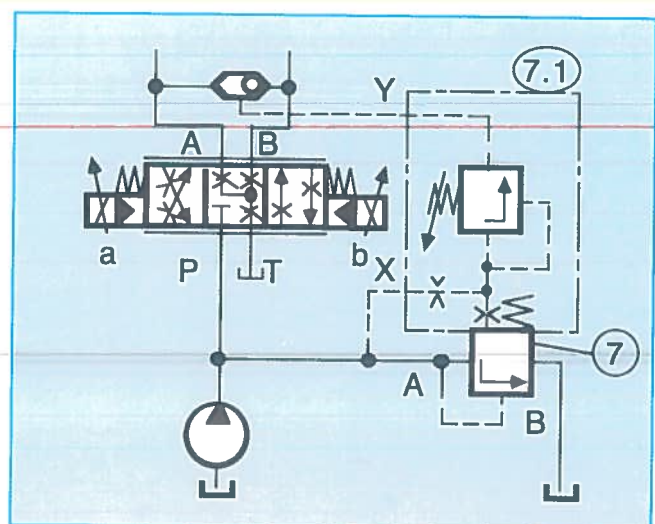


Fig. 32: 3-way pressure compensator  $\Delta p$  variable



Fig.	size Item	16	25	32	40	50	63
21	1	LC16DR80D60	LC25DR80D60	LC32DR80D60	LC40DR80D60	LC50DR80D60	LC63DR80D60
	1.1	LFA16D8-60	LFA25D8-60	LFA32D8-60	LFA40D8-60	LFA50D8-60	LFA63D8-60
22	2	LC16DR80D60	LC25DR80D60	LC32DR80D60	LC40DR80D60	LC50DR80D60	LC63DR80D60
	2.1	LFA16D17-60	LFA25D17-60	LFA32D17-60	LFA40D17-60	LFA50D17-60	LFA63D17-60
23	3	LC16DR80D60/A07	LC25DR80D60/A08	LC32DR80D60/A08	LC40DR80D60/A10	LC50DR80D60/A12	LC63DR80D60/A15
	3.1	LFA16D8-60	LFA25D8-60	LFA32D8-60	LFA40D8-60	LFA50D8-60	LFA63D8-60
24	4	LC16DR40D60	LC25DR40D60	LC32DR40D60	LC40DR40D60	LC50DR40D60	LC63DR40D60
	4.1	LFA16DB2-60/050	LFA25DB2-60/050	LFA32DB2-60/050	LFA40DB2-60/050	LFA50DB2-60/050	LFA63DB2-60/050
25	5	LC16DR40D60	LC25DR40D60	LC32DR40D60	LC40DR40D60	LC50DR40D60	LC63DR40D60
	5.1	LFA16DB2-60/050	LFA25DB2-60/050	LFA32DB2-60/050	LFA40DB2-60/050	LFA50DB2-60/050	LFA63DB2-60/050
29	6	LC16DB80D60	LC25DB80D60	LC32DB80D60	LC40DB80D60	LC50DB80D60	LC63DB80D60
	6.1	LFA16D8-60	LFA25D8-60	LFA32D8-60	LFA40D8-60	LFA50D8-60	LFA63D8-60
30	7	LC16DB40D60	LC25DB40D60	LC32DB40D60	LC40DB40D60	LC50DB40D60	LC63DB40D60
	7.1	LFA16DB2-60/050	LFA25DB2-60/050	LFA32DB2-60/050	LFA40DB2-60/050	LFA50DB2-60/050	LFA63DB2-60/050
31	8	LC16DB40D60	LC25DB40D60	LC32DB40D60	LC40DB40D60	LC50DB40D60	LC63DB40D60
	8.1	LFA16DBU2K.-60/.	LFA25DBU2K.-60/.	LFA32DBU2K.-60/.	LFA40DBU2K.-60/.	LFA50DBU2K.-60/.	LFA63DBU2K.-60/.
Q <sub>max</sub>	8 bar spring	75 L/min for $\Delta p = 5$ bar	150 L/min for $\Delta p = 5$ bar	250 L/min for $\Delta p = 5$ bar	500 L/min for $\Delta p = 5$ bar	550 L/min for $\Delta p = 5$ bar	850 L/min for $\Delta p = 5$ bar

Fig. 34 List of equipment available

## Load Compensation with 2-way Cartridge Valves

### Aids in project engineering for the correct selection of the size for logic elements

If pressure reducing logic elements are used as a pressure compensator for flow control, the characteristic curves for the pressure reducing function specified in the data sheet cannot be used for selection. The following observation indicates the selection criteria and their derivation for this particular application.

#### Power Limit for Pressure Control

In the case of the pressure reducing function, the pilot pressure for the spring side is obtained directly at the output of the element (see Fig. 36).

The power limit is reached when the spring force is compensated by the impulse forces of the flow. While neglecting the unsteady proportion, the axial components of this impulse force for the control volume shown in Fig. 35 is obtained from the following relationship:

$$F_{ax} = \varphi \cdot Q (\omega_E \cdot \cos \alpha + \omega_A)$$

where

$F_{ax}$  = Force in axial direction

$\varphi$  = Density of the flowing medium

$Q$  = Volumetric flow

$\omega_E, \omega_A$  = Inlet and outlet speed

$\alpha$  = Inlet angle

In this case, the calculation of  $F_{ax}$  involves considerable difficulties since the angle  $\alpha$  is difficult to determine accurately due to the relatively complicated geometry of the control land (holes plus fine control grooves), this also applies to the outlet speed  $\omega_A$  due to the short distance between the point of deflection and outlet from the control volume.

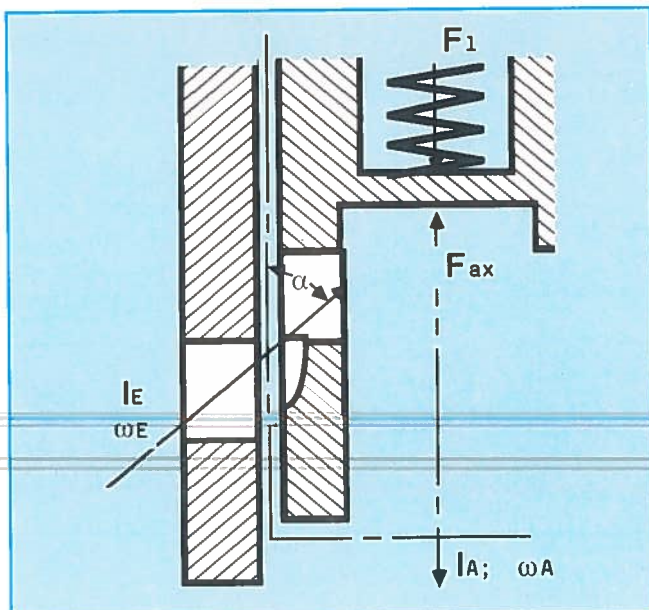


Fig. 35

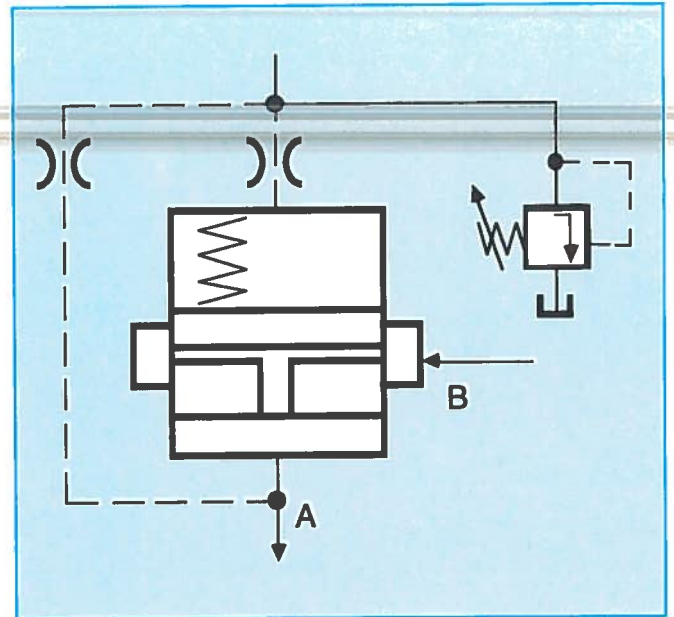


Fig. 36

However, it is relatively simple to determine  $F_{ax}$  by experiment.

The spring pretension  $F_1$  is known.

If  $F_{ax}$  is greater than  $F_1$ , the piston moves in the closed direction. Thus, for the pressure reducing logic elements, this point is reached when the flow can no longer be increased. It is thus a function of  $\Delta p$ .

#### Power Limit for Flow Control

If the logic elements are used as a pressure compensator for flow control, the pressure for the spring chamber is obtained after the control orifice (proportional valve) (Fig. 37). The power limit for flow control is reached when the sum of the previously described impulse forces  $F_{ax}$ ,  $\Delta p_{BI}$  of the orifice and the  $\Delta p_L$  of the connecting line balances the spring force  $F_1$ .

$$F_1 = F_{ax} + \Delta p_{BI} \cdot A_K + \Delta p_L \cdot A_K$$

$A_K$  = Spool area

The diagrams show the previously specified relationships for the sizes 32 and 40 (Figs. 38 and 40). The horizontal lines represent the spring pretension  $F_1$  independent of flow and referred to the relevant spool area  $A_K$  in the form of  $\Delta p$ .

$$F_1/A_K = \text{constant}$$

These lines are terminated at the maximum flow rates determined from the pressure reduction measurements, at which the spring forces are compensated by the flow forces. The connecting lines of these termination points are represented by the following function

$$F_{ax}/A_K = f(Q).$$

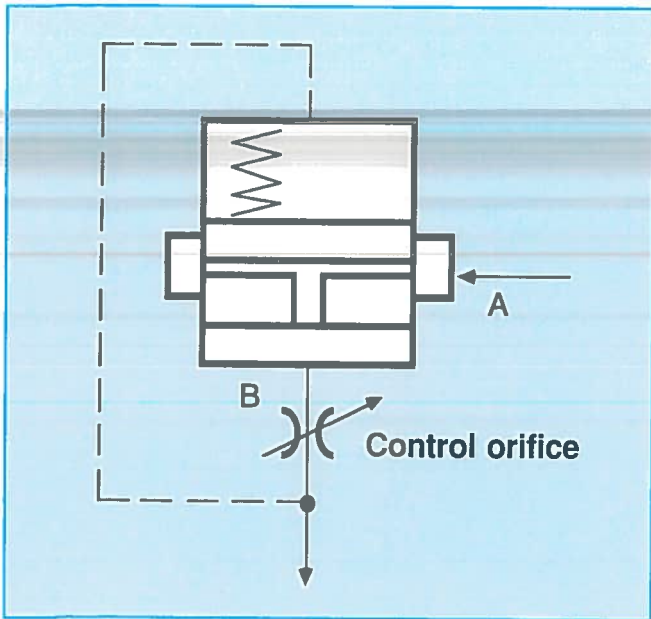


Fig. 37

The pressure difference

$$\Delta p_{BL} + \Delta p_L = (F_1 - F_{ax})/A_K$$

available across the orifice and associated throttle system, can be read off for each spring referred to set maximum volume as the vertical distance between the two curves

$$F_1/A_K = \text{constant and } F_{ax}/A_K = f(Q).$$

## Example

The load in a control for  $Q = 340 \text{ l/min}$  is to be compensated with the aid of a pressure reducing logic element.

A valve type 4 WRZ 32 E 360 is used, i.e. 360 l/min at 10 bar, total valve pressure drop. Therefore 5 bar  $\Delta p$  per control land is available for 340 l/min, so that the following  $\Delta p$  is necessary at the control edge

$$Q = Q_N \cdot \sqrt{\Delta p / \Delta p_N}$$

$$\Delta p = (Q/Q_N)^2 \cdot \Delta p_N$$

$$\Delta p = (340/360)^2 \cdot 5 = 4,45 \text{ bar} \approx 5 \text{ bar}$$

$Q_N$  = Nominal flow of valve

$\Delta p_N$  = Nominal  $\Delta p$  of the valve

$\Delta p$  = Required  $\Delta p$

The correct logic element can be selected with the aid of the characteristic curves. In the case of the logic element LC32 DR 80, a  $\Delta p$  of only approx. 3 bar would be available for the valve at 340 l/min, i.e. the  $\Delta p$  at the valve would be too low to guarantee the required flow.

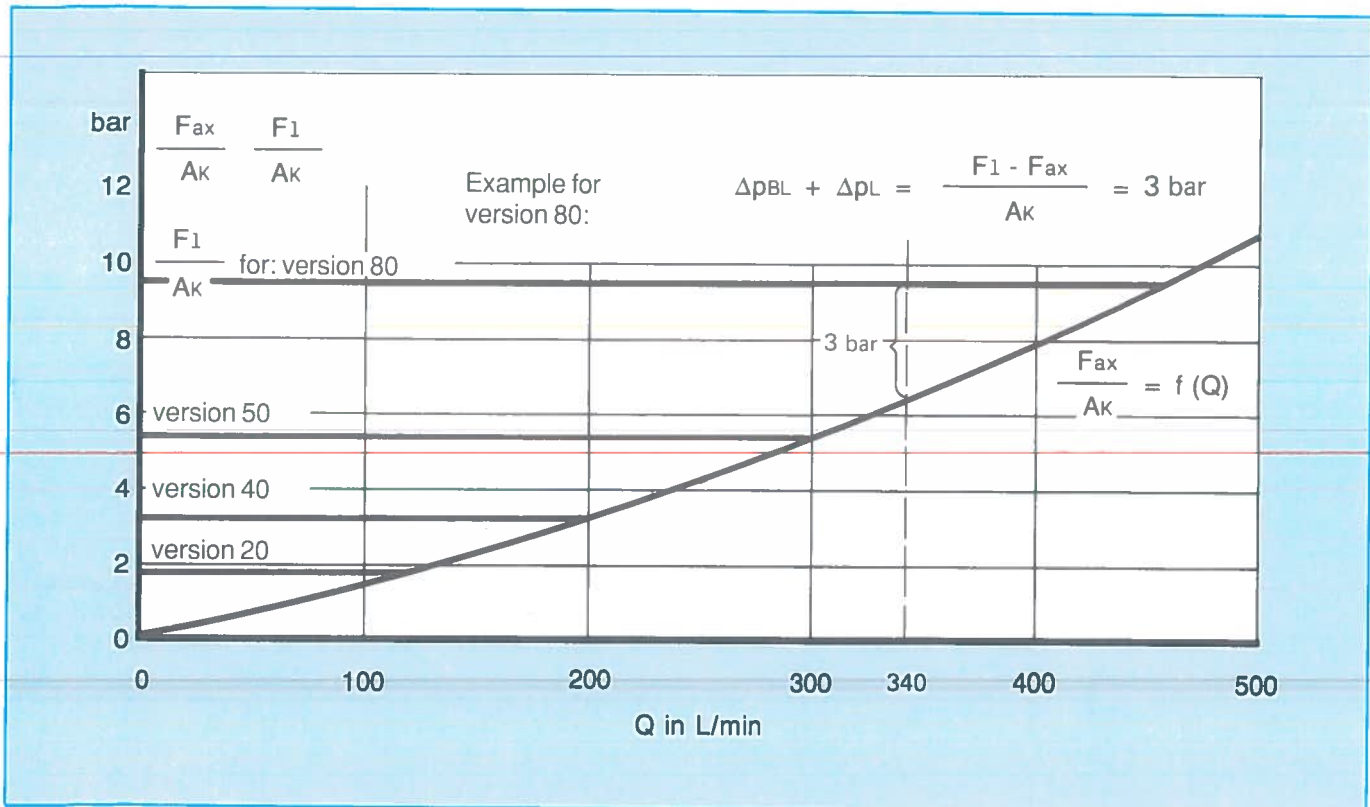


Fig. 38 Power limit for 2-way cartridge valve, size 32

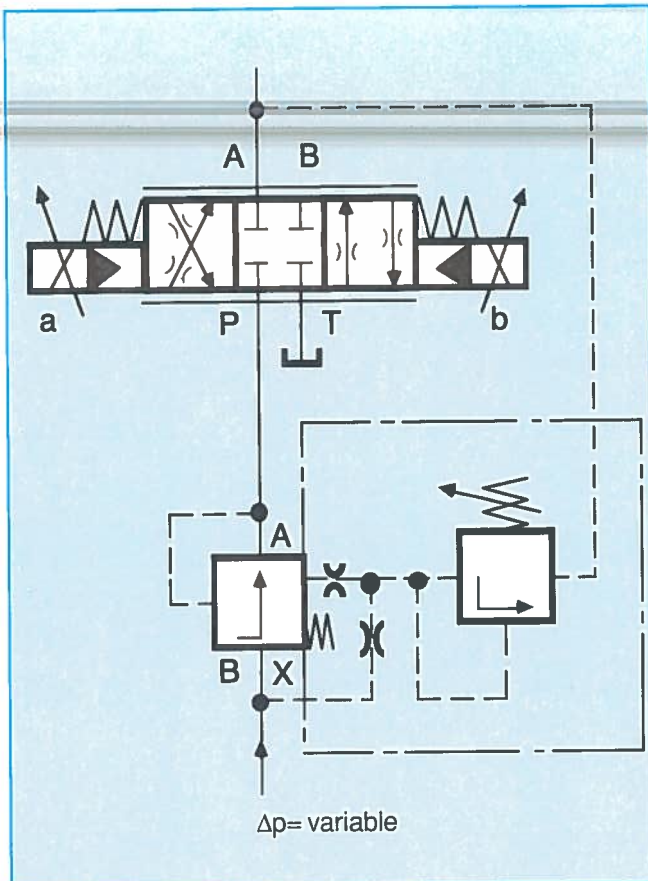


Fig. 39 Pressure compensator with variable  $\Delta p$

It is now possible to increase the  $\Delta p$  by changing the circuit (see Fig. 39). The version LC 32 DR 40 (with 4 bar spring) should however, be used in this case.

The other alternative would be to select a larger logic element LC 40 DR 80. At  $Q = 340$  l/min, this permits a  $\Delta p$  of 7 bar at the valve throttle edge.

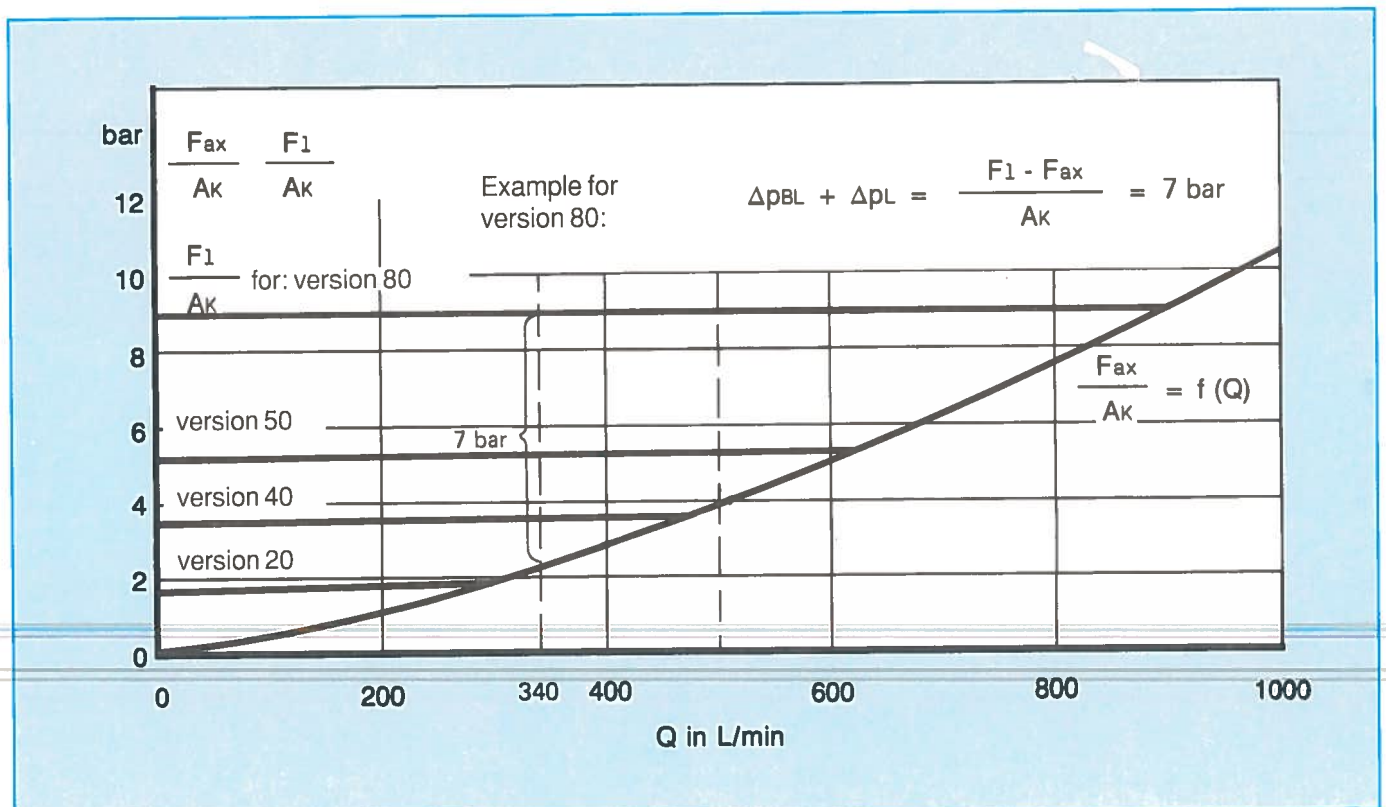


Fig. 40 Power limit for 2-way built-in valve, size 40

Notes

Chapter D

# **Electronic Controls for Proportional Valves**

Heribert Dörr



## Definitions and Explanations

In this chapter, the most important components of the electronic controls for proportional valves are explained together with terms, functions and block diagrams. This is intended as an aid for those who previously have had little or nothing to do with this subject.

### Ramp Generator

The ramp generator produces from a stepped signal value as the input signal a slowly raising or falling output signal. The change of the output signal with time can be varied by means of a potentiometer.

The operating principle of the ramp generator is based on the concept that the capacitor  $C$  can only be charged at the rate allowed by the potentiometer so that the output voltage constantly changes slowly with respect to a stepped input signal.

The increase in the output voltage can be influenced via the variable resistor  $R$ , thereby determining the charge rate of the capacitor.

The set ramp time is always referred to 100 % signal value (stepped input signal).

#### Example

Set ramp time of max. 5 seconds at 100 % signal value: If, for example a signal value of 60 % is set then the signal value is reached after approx. 3 seconds.

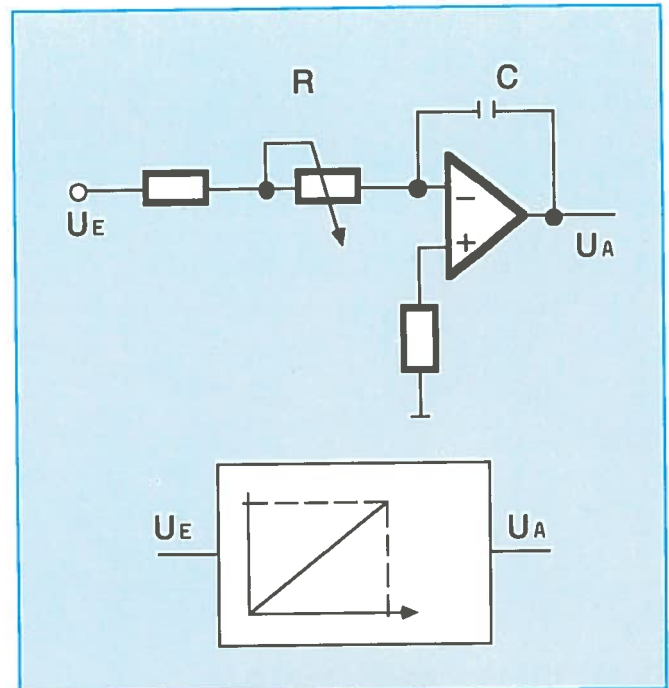


Fig. 1 Ramp generator

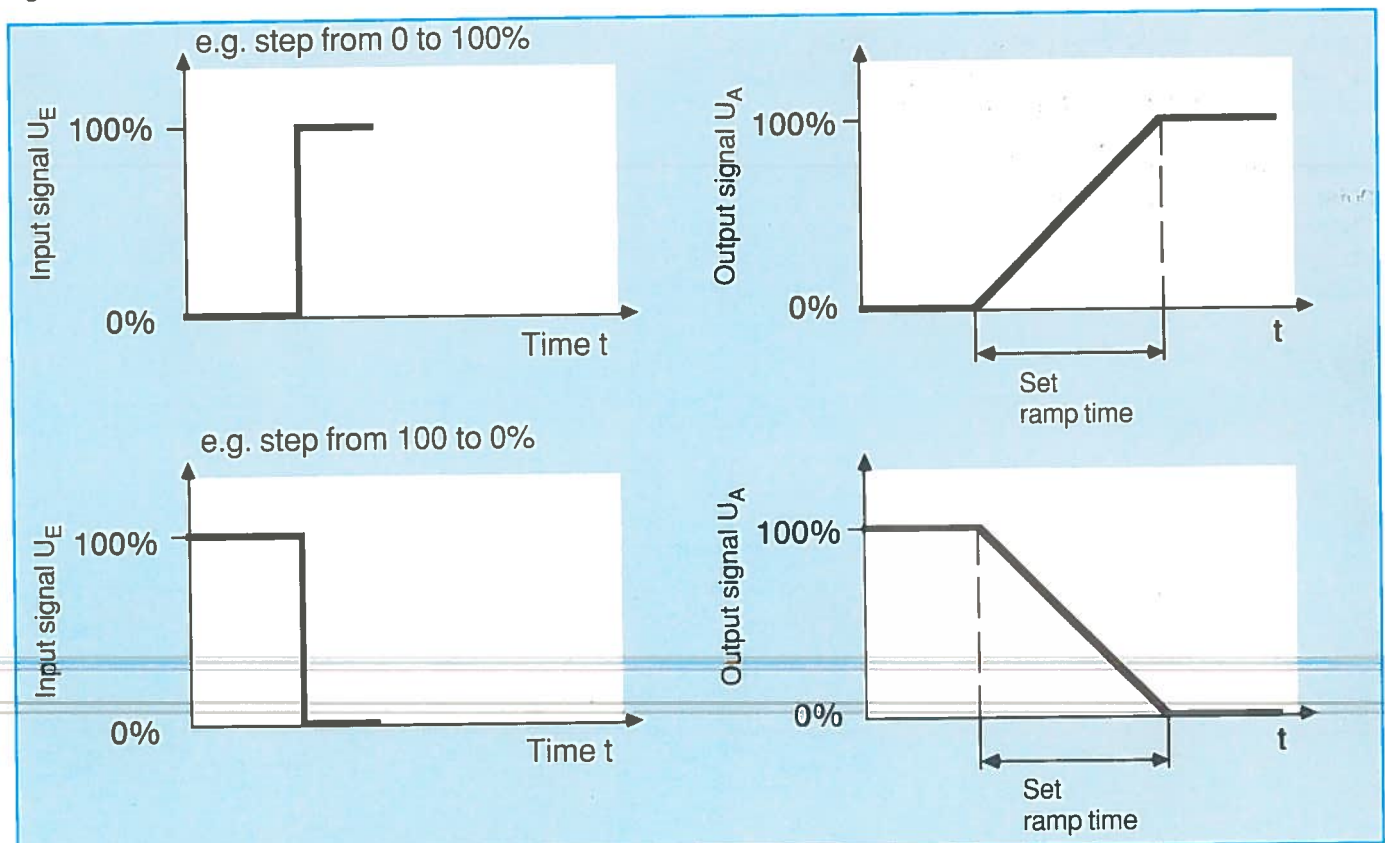


Fig. 2 Stepped signal, ramp time

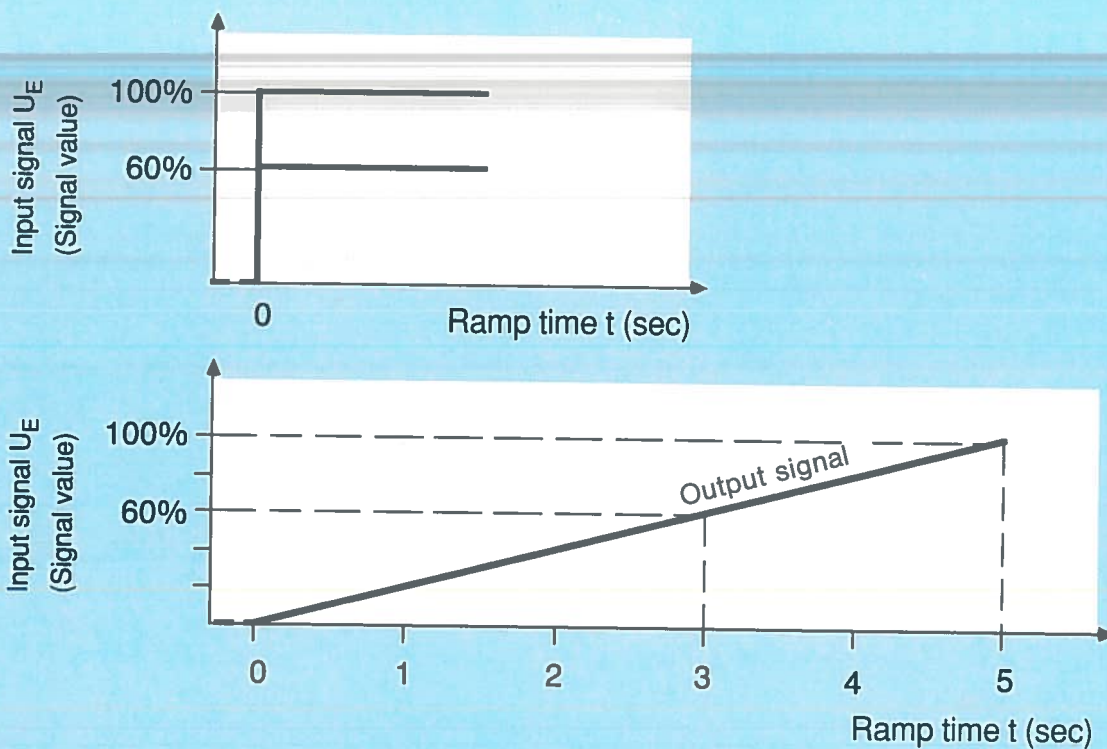


Fig. 3 Ramp time as function of input signal

## Pulse Width Modulated Output Stage

The signal value voltage is converted into a solenoid current in the output stage.

The solenoid current is pulsed in order to maintain the power loss of the output stage and therefore the thermal load of the pc board as low as possible.

The pulse frequency is determined with the clock pulse generator dependent on the type of valve.

The effective current to the solenoid is dependent upon the on/off time ratio of the output power transistor.

### Example

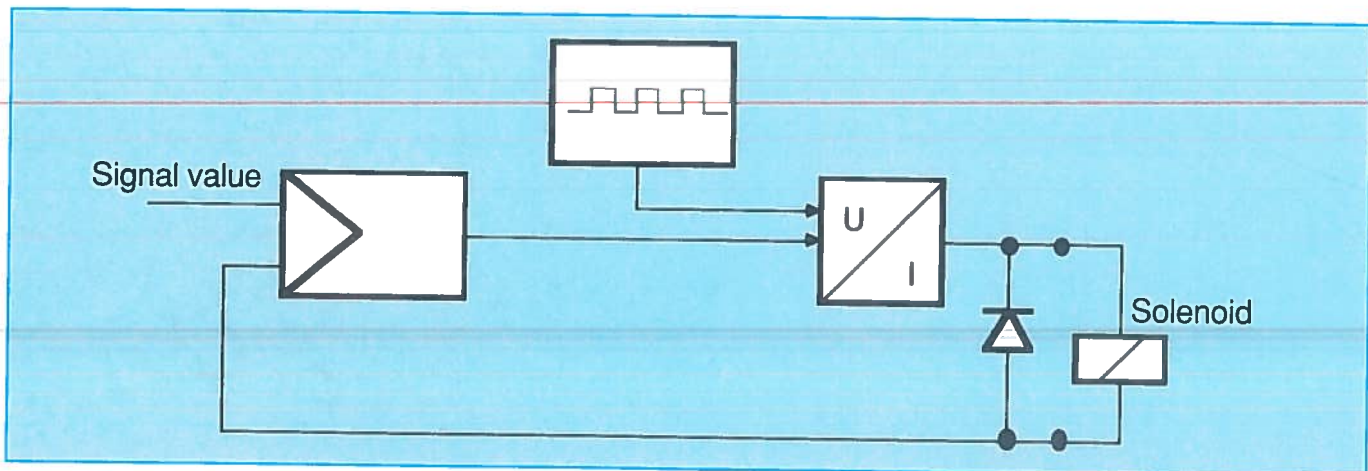
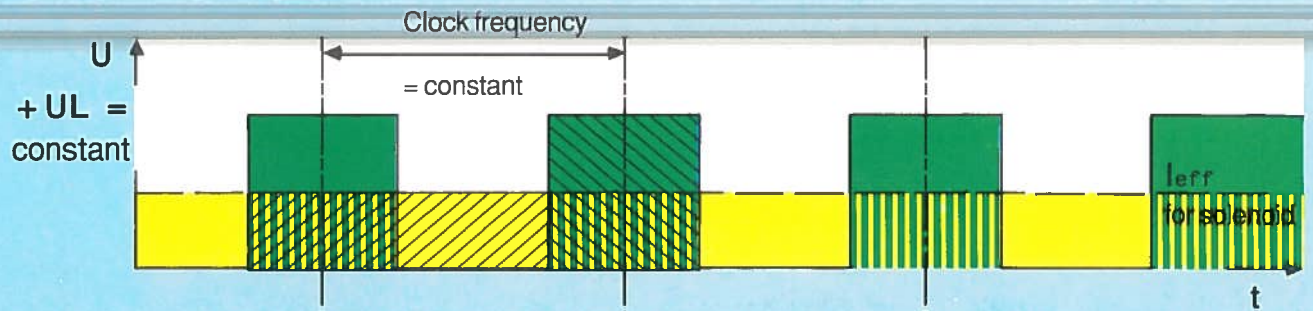


Fig. 4 Pulse width modulated output stage

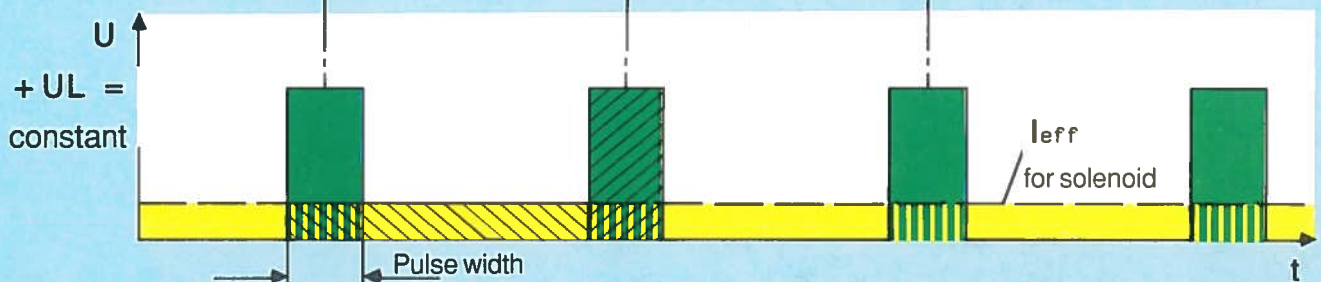
## a) Output stage fully "on"



Area corresponds to area resulting in  $i_{eff}$  for solenoid valve.

The pulsed voltage produces a certain current  $i_{eff}$  over the resistance of the solenoid.

## b) Output stage partly "on"



Area corresponds to Area resulting in  $i_{eff}$  for the solenoid.

The pulse width (i.e. the area under the pulses), produces an equivalent effective output current.

Fig. 5 Pulse width modulated output stage fully and partly "on".

## Power Supply

The power supply for all proportional amplifier cards can be formed as shown in Fig. 6.

To increase reliability, 2 terminals are always used for connection of the power supply (Fig. 7).

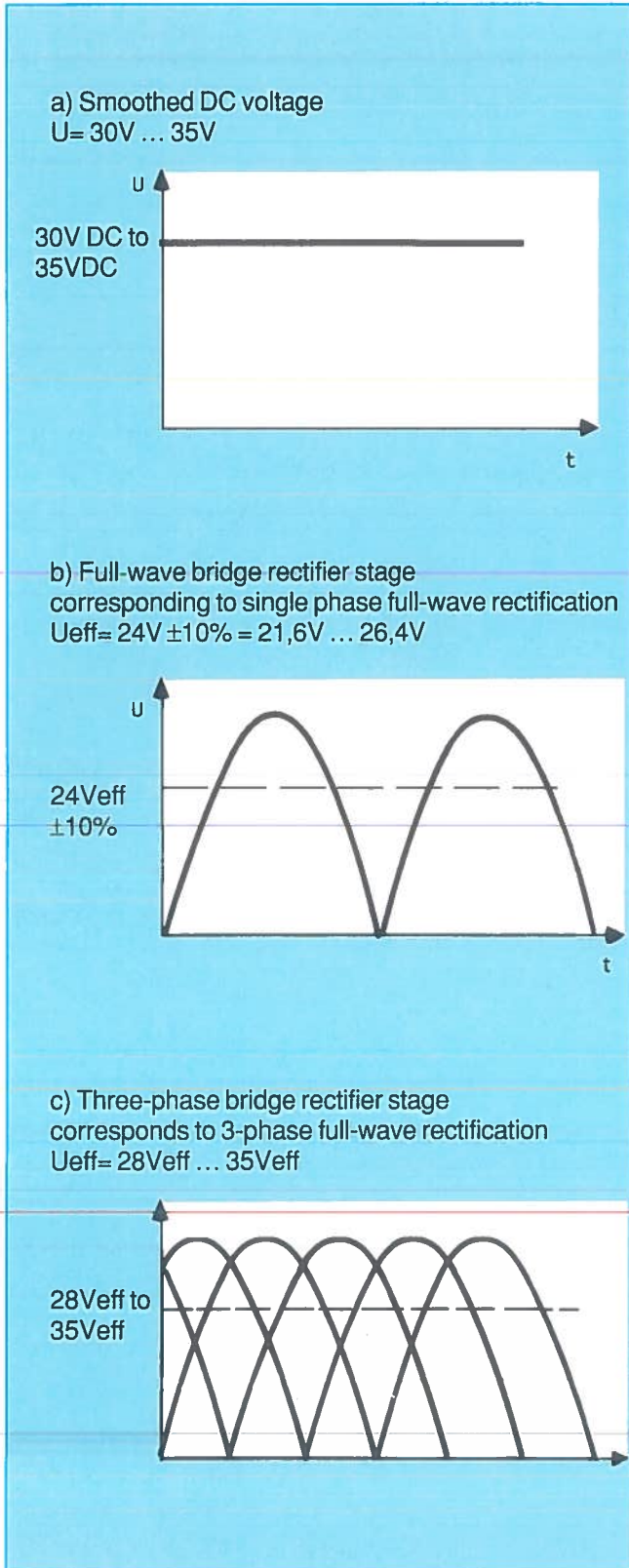


Fig. 6 Power supply

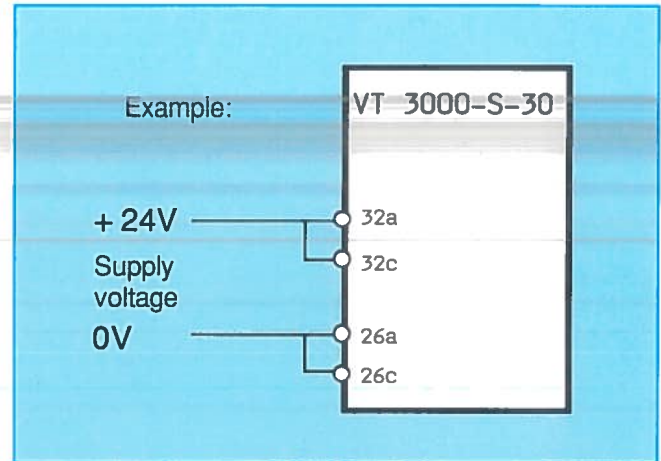


Fig. 7

### Example

Configuration of the voltage supply stages on the amplifier cards based on the example of the single-phase full-wave rectifier stage (Fig. 8).

Conversion of the voltage available from the consumer network takes place in the first section from 220 V AC to 24V DC. It is then fed to the amplifier card.

The input voltage is smoothed in the 2nd section.

Conversion of the smoothed voltage into a stabilized voltage of 18V takes place in the 3rd section. By selecting a new reference point M0, the stabilized voltage of  $\pm 9V$  is obtained referred to this point M0.

The following points must be observed for all amplifier cards

- The power supply must be disconnected before unplugging the amplifier card.
- Measurements on DC voltage scale only.
- Test point (M0) is raised by + 9 V with respect to 0 V supply voltage.
- Do not connect M0 to 0 V supply voltage.
- Do not connect the ground symbol on the inductive positional transducer to the 0 V supply voltage.
- A minimum distance of 1 m must be maintained from radio equipment.
- Signal values must be switched only by means of contacts, suitable for currents 1 mA.
- Signal value lines and lines of the inductive positional transducer must be screened. Screen to be open on one end: connect card end to 0 V supply voltage.
- Do not lay solenoid lines in the vicinity of power carrying lines.

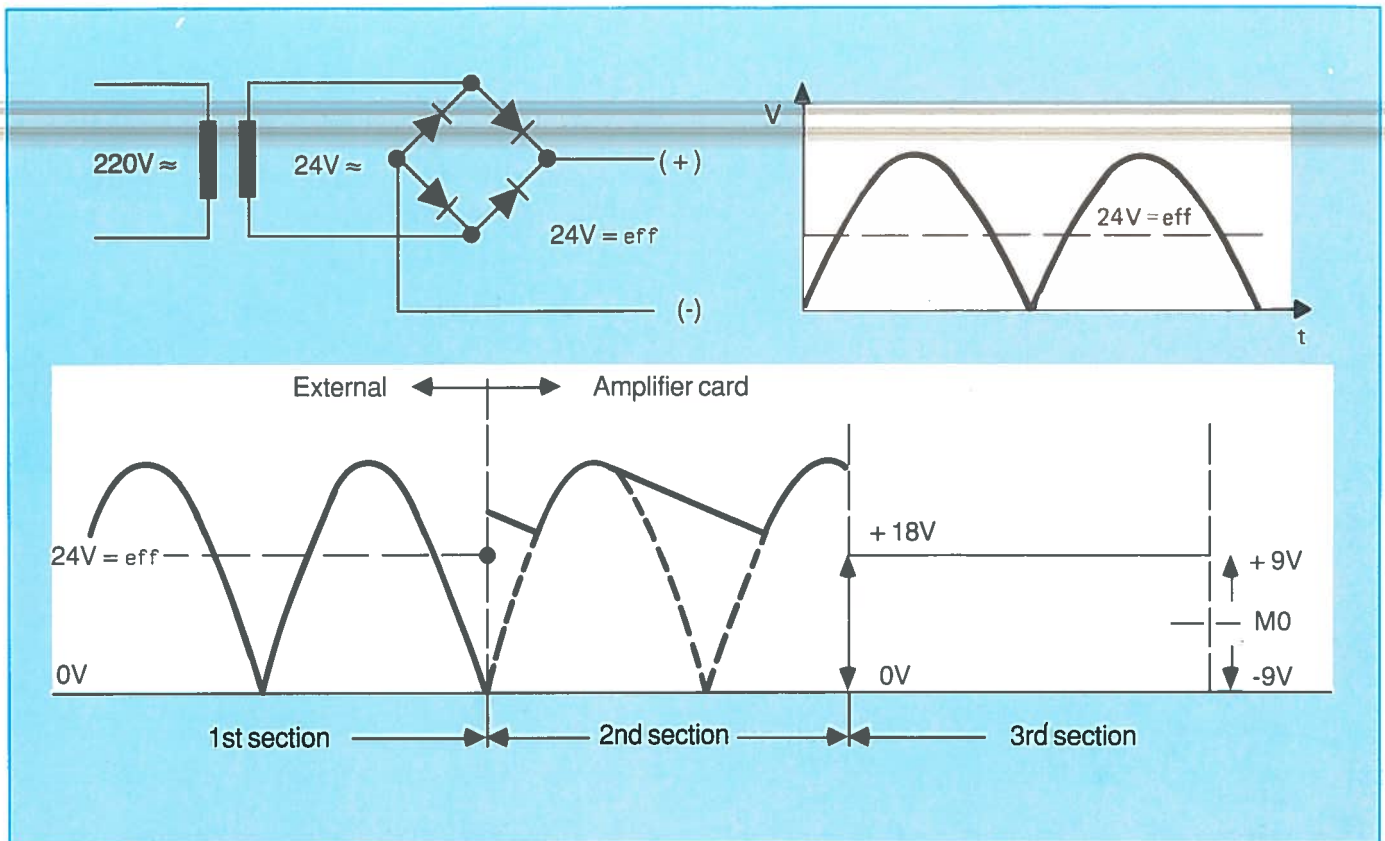


Fig. 8 Singlephase full-wave rectifier stage

### Cable Break Detection

The cable break detection facility monitors the supply line to the transducer. In the case of fault, i.e. one of the three cores of the connecting cable for the positional transducer is broken, the power to both solenoids A and B is cut and the valve assumes its mid-position.

### Step Function Generator

The step function generator produces a constant output signal at signal value voltages greater than 100 mV. The output signal is 0 V at signal value voltages less than 100 mV.

The output signal of the function generator causes a current step at the solenoid. This current step serves the purpose of quickly overcoming the positive overlap of the proportional valves.

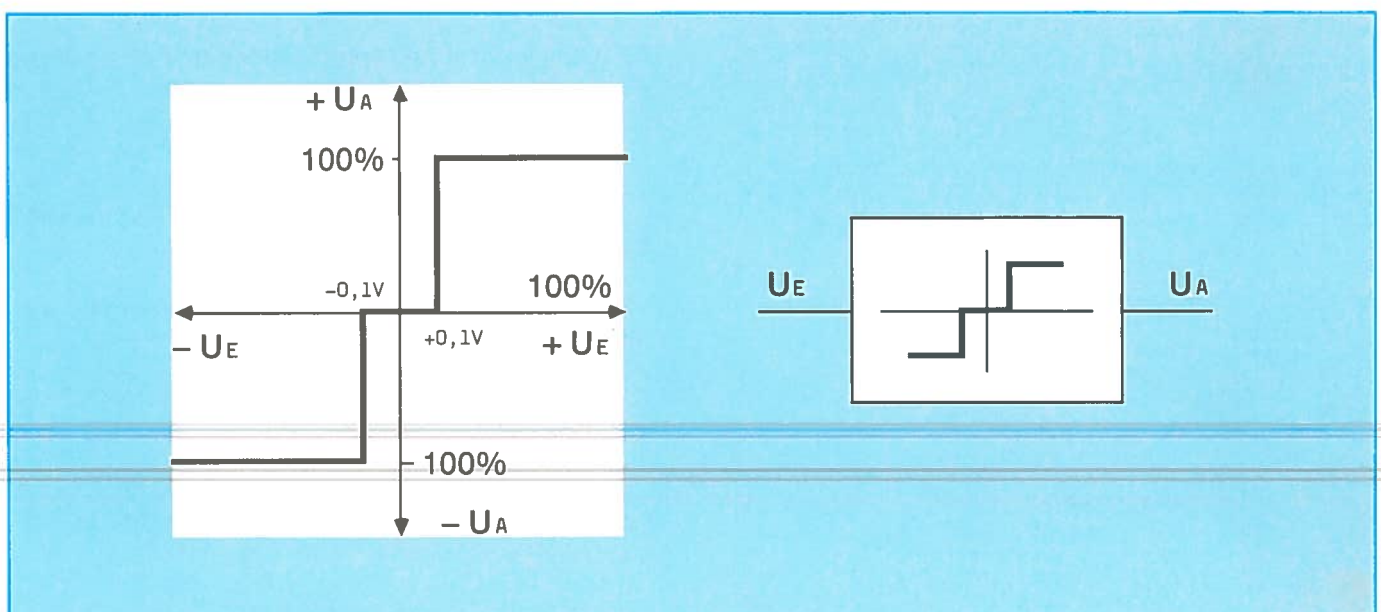


Fig. 9 Step function generator

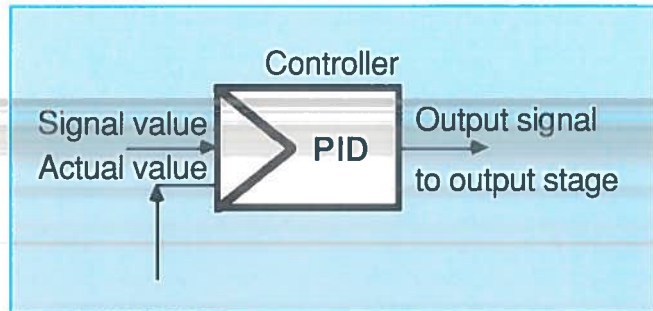


Fig. 10 PID controller

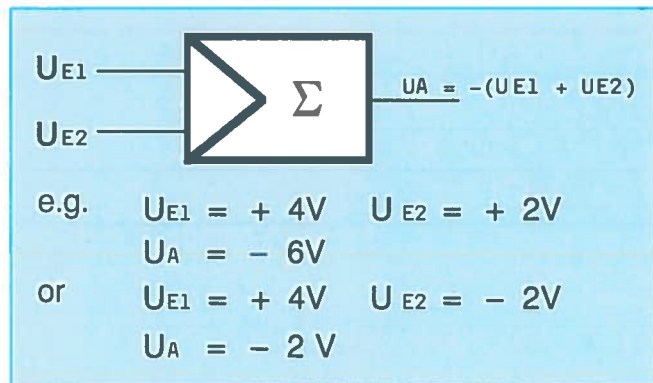


Fig. 11 Summator

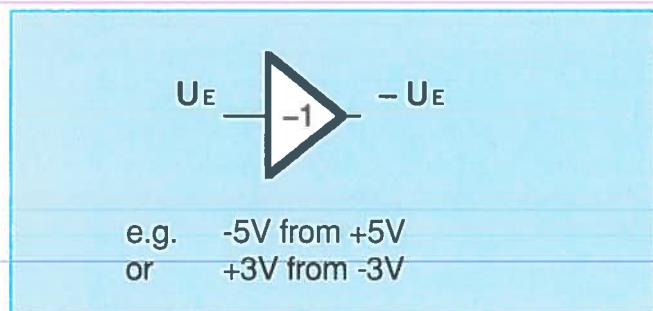
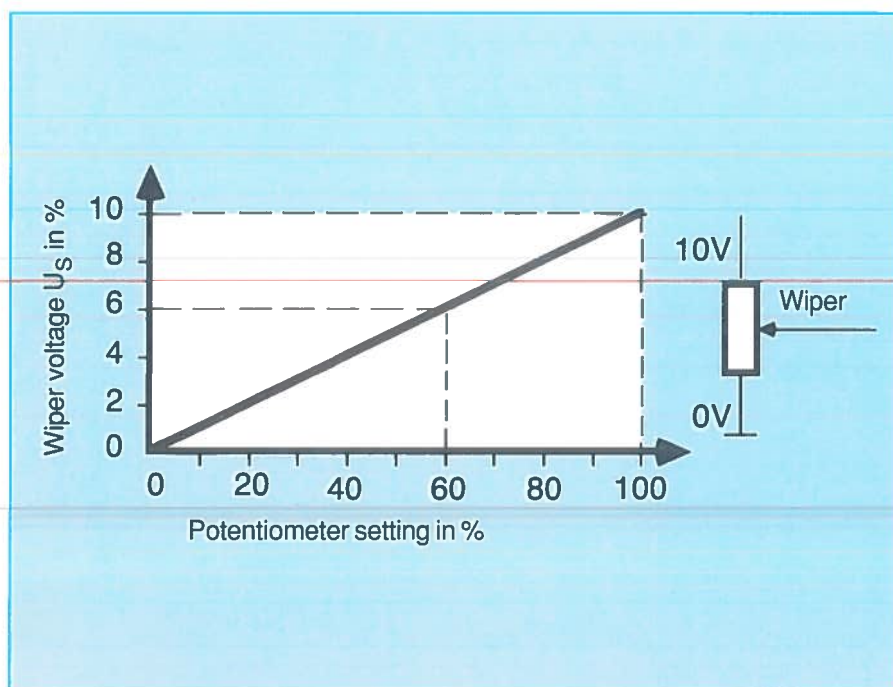


Fig. 12 Inverter



## Potentiometer

The potentiometer is an ohmic resistor with a variable wiper.

If the ends of the potentiometer are connected to 0 V and 10 V, any value between 0 ... 10 V can be obtained at the wiper.

### Example

At a setting of 60 %, a voltage of 6 V is available at the wiper.

Fig. 13 Potentiometer

## Controller on Proportional Amplifier Cards

The controllers of the proportional amplifier cards are specially adapted to the various types of valve. Corresponding to the difference signal value - feedback value, the controller produces an output signal which controls the pulsed output stage.

## Summator

The summators on the proportional amplifier cards add two voltages, whereby the addition signal is inverted.

## Inverter

The purpose of the inverters on the proportional amplifier cards is to reverse the polarity of the input voltage.

### Pilot Current

The pilot current is a solenoid current. The pilot current of the solenoid is applied as soon as the amplifier card is connected to the supply voltage and the valve on the amplifier is also connected. It is used for maintaining the pulse frequency, for premagnetization of the solenoid and ensures the solenoid of the valve starts quickly from its initial position when a signal value is given.

### Inductive Positional Transducer on the Valves

The inductive positional transducer serves the purpose of contactless measurement of the spool stroke.

The inductive positional transducer consists of a cylindrical body carrying the coils in which a measuring armature with a ferromagnetic core is immersed.

The sensor consists of two coils which are connected such as to form an inductive centre tapped coil.

The inductive positional transducer is fed with a carrier frequency of 2.5 kHz. The amplitude of this carrier frequency varies at the output depending on the position of the measuring armature. The inductance of the coils varies as the measuring armature is shifted.

$$Z_{\omega} = R + j\omega l$$

( $j\omega l$  = Reactance)

In conjunction with the inductance, the AC resistance  $Z_{\omega}$  also changes and therefore the output amplitude of the frequency.

The output amplitude is equivalent to  $U_S$  when the measuring armature is in the centre position. If the measuring armature is deflected, the output amplitude (Fig. 15) shifts in direction  $U_{S1}$  or  $U_{S2}$ .

The demodulator converts the output amplitude into a corresponding DC signal.

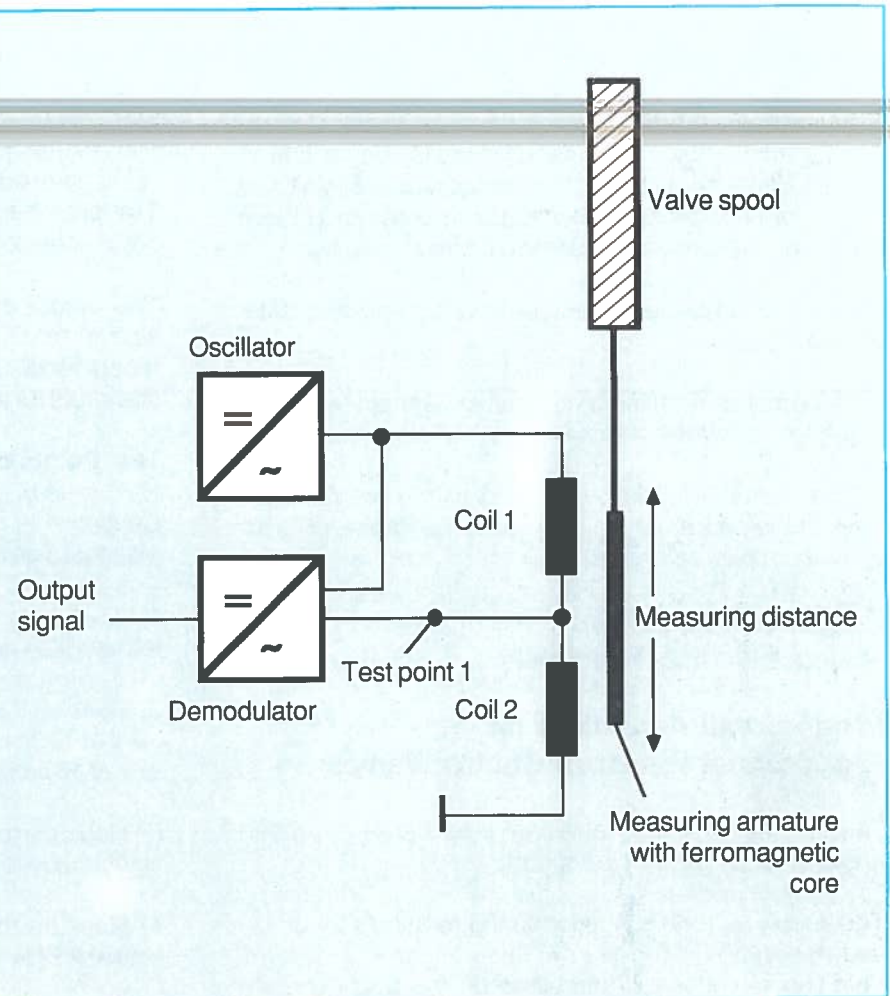


Fig. 14 Schematic diagram of inductive positional transducer

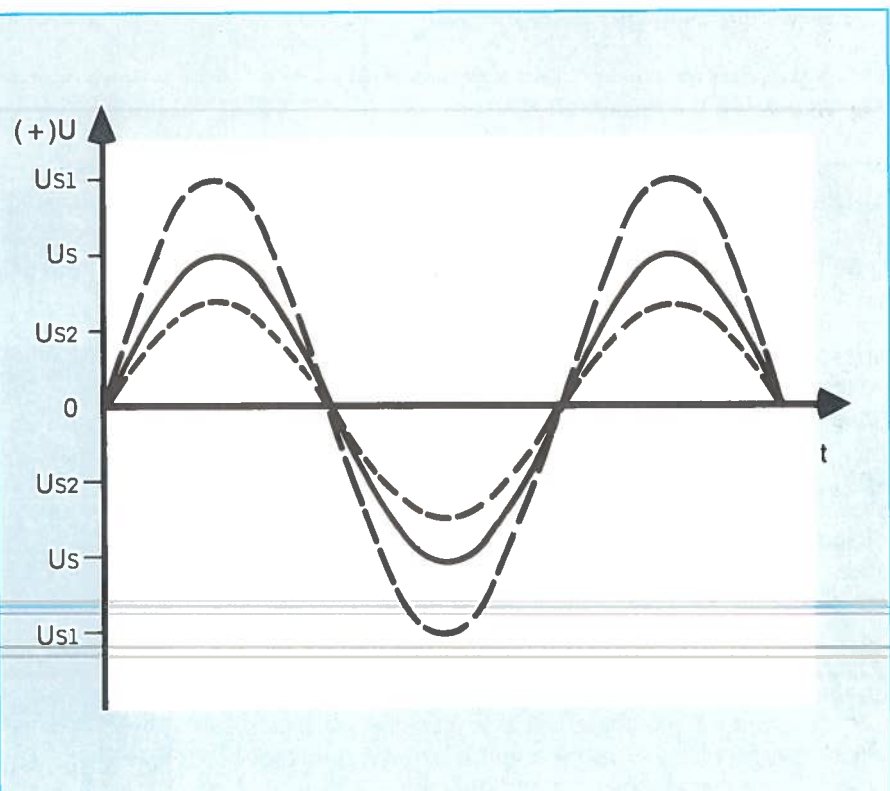


Fig. 15 Output amplitude measured at test point 1 corresponding to Fig. 14

### Proportional Amplifiers for Proportional Valves

Euro-format 100x160 mm electronic circuit boards have been developed and standardized for the various proportional valves. Each amplifier card is assigned to a certain type of proportional device in order to achieve optimum adaptation and therefore optimum results.

The proportional amplifiers are divided into two categories:

- Proportional amplifiers for valves without electrical feedback (for force-controlled proportional solenoids)
- Proportional amplifiers for valves with electrical feedback of the proportional valve spool (for stroke-controlled proportional solenoids).

#### Proportional Amplifiers without Electrical Feedback

### Proportional Amplifiers for Proportional Pressure Control Valves

The function of the proportional amplifier is described based on the given block diagram.

The supply voltage is applied to the terminals 24 ac (+) and 18 ac (0 V).

This supply voltage is smoothed on the amplifier card (1) and a stabilized voltage of  $\pm 9$  V is derived from this smoothed voltage.

The stabilized voltage of  $\pm 9$  V is used for:

- a) The supply of external or internal potentiometers. The voltage + 9 V is tapped off at 10 ac - 9 V at 16 ac.
- b) The supply of the internal operational amplifiers.

A potentiometer R2 for signal value setting is mounted on the amplifier card. In order to set a signal value voltage at R2, the stabilized voltage of + 9 V must be applied to the signal input 12 ac. The signal value voltage tapped off at the potentiometer R2 is fed to the ramp generator (2).

The ramp generator (2) generates from a stepped signal a slowly raising or falling output signal. The slope of the rise of the output signal, i.e. the change with time, can be set via the potentiometers R3 (for up ramp) and R4 (for down ramp).

The specified ramp time of max. 5 seconds, can only be reached over the entire voltage range (from 0 V to + 6 V) measured at the signal value test sockets. A signal value voltage of + 9 V at the input produces a voltage of + 6 V at the signal value test sockets.

The output signal of the ramp generator is fed to the pulsed output stage (3), as well as the voltage signal of the potentiometer R1.

The pilot current for the proportional solenoid can be set at potentiometer R1.

The output stage (3) actuates the proportional solenoid at max. 800 mA. The current flowing through the proportional solenoid can be measured at the test socket X2 as a DC voltage ( $1V = 1A$ ).

#### Test Points on Proportional Amplifier

##### Caution

Measured on DC voltage setting.

- 1) Measure the supply voltage of + 24 V at the terminals 24 ac with respect to 18 ac.
- 2) Measure the stabilized voltage  $\pm 9$  V  
+9 V at 10 ac with respect to 14 ac  
-9 V at 16 ac with respect to 14 ac
- 3) Measure the signal value voltage of 0 to +6 V at the signal value test socket X1
- 4) Measure the solenoid current of 0 to 800 mA at the socket X2 ( $1V = 1A$ )

#### Control Example

The following connection assignments remain constant:

- Valve connection at 22 ac and 20 ac
- Supply voltage 24 V at 24 ac (+) and 18 ac (-)

#### Function

- Remote setting via potentiometer with call via relay
- Remote setting via differential input
- Switch off external ramp up and down

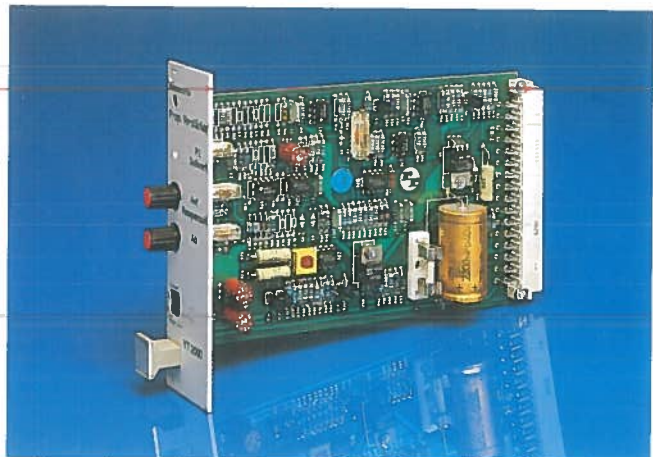


Fig. 16 Proportional amplifier Type VT 2000 S 40

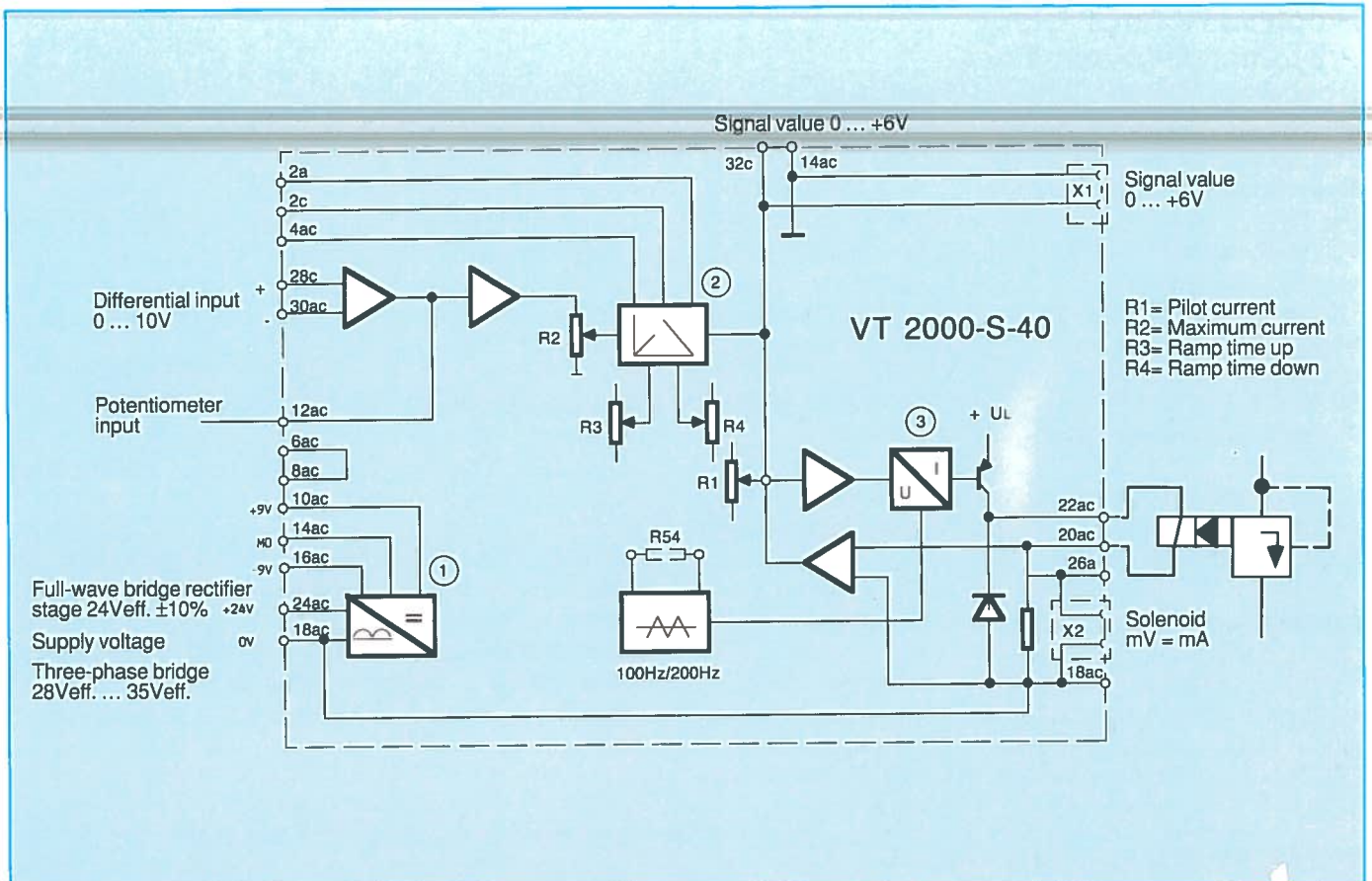


Fig. 17 Terminal connections proportional amplifier VT 2000 S 40

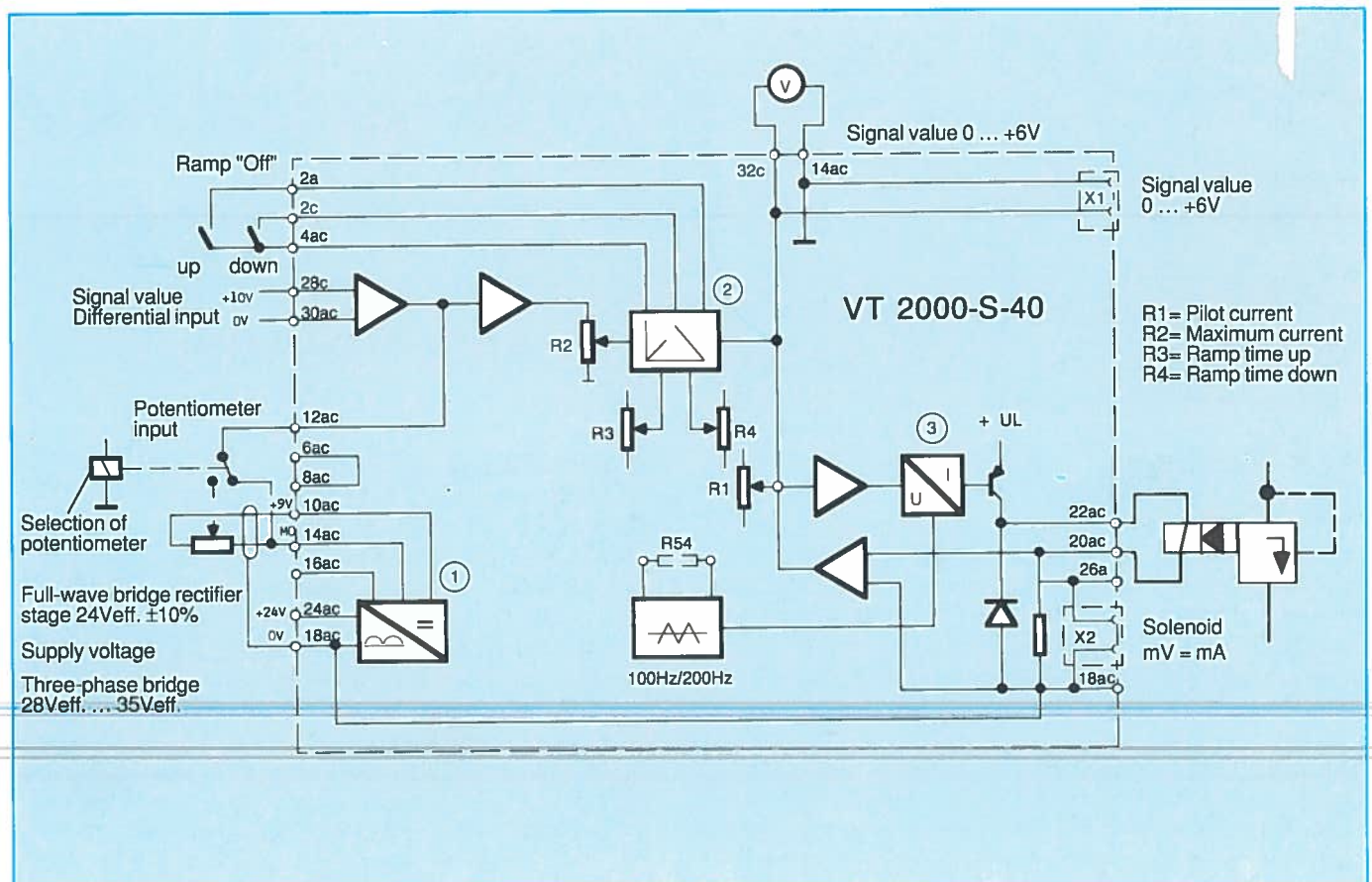


Fig. 18 Control example with proportional amplifier Type VT 2000 S 40

### Proportional Amplifier for Pilot Operated Proportional Directional Valves without Feedback

The function of the proportional amplifier is described based on the given block diagram.

The proportional amplifier receives its voltage supply via the terminals 32 ac (+) and 26 ac (0 V). The supply voltage is smoothed on the amplifier card (7) while at the same time being used to form a stabilized voltage of  $\pm 9$  V.

The stabilized voltage  $\pm 9$  V is used

a) for supplying the external potentiometers or the internal potentiometers and can be tapped off at 20 ac (+ 9 V) and at 26 ac (- 9 V).

b) for supplying the internal operational amplifiers.

4 potentiometers are mounted on the amplifier card for setting the signal values R1 to R4 (8). In order to set a signal value voltage, the 4 terminals of the signal inputs 12a, 8a, 10a, 10c must be connected to the stabilized voltage of + 9 V terminal 20c or - 9 V terminal 26ac.

The solenoid B is active when the signal inputs are applied to + 9 V. The solenoid B is connected to the terminals 22a and 28a.

The solenoid A is active when the signal inputs are applied to - 9 V. Solenoid A is connected to the terminals 30a and 24a.

The set signal value voltages R1 to R4 are selected via the relays K1 to K4.

The selection voltage of the relays is available at 28c and can be applied via potential-free contacts to the relay inputs 8c, 4a, 6a, 6c.

A voltage signal is generated at the input of the ramp generator (1) when the signal value potentiometers R1...R4 are selected.

The ramp generator (1) generates from a stepped rising input signal a slowly rising output signal. The rise time (slope) of the output signal can be varied with the potentiometer R8 (ramp time). The specified ramp time of max. 5 sec. can only be reached over the entire voltage range from 0 V to  $\pm 6$  V, measured at the signal value test sockets.

The signal value voltage  $\pm 9$  V at the input results in a voltage of  $\pm 6$  V at the signal value test sockets. The maximum ramp time is shortened if a lower signal value than  $\pm 9$  V is switched to the input of the ramp generator (1).

The output signal of the ramp generator (1) is fed to the summator (3) and the step function generator (2). The step function generator (2) generates at its output a step function which is added in the summator (3) to the output signal of the ramp generator (1). The step function is required to move through the neutral point overlap of the valve quickly.

This step is effective as of low signal value voltages (less than 100 mV). The step function generator (2) generates a constant signal if the signal value voltage increases to a higher value.

The output signal of the summator (3) acts on the two output stages with current regulator (4), pulse generator (5) and power amplifier (6). The output stage for solenoid B is actuated in the case of a positive signal value voltage at the input of the amplifier. The output stage for solenoid A in the case of a negative signal value voltage.

The following should also be noted:

a) Differential signal input from 0 to  $\pm 10$  V

This input is required in order to achieve high impedance isolation between the valve amplifier card and an external electronic control system.

b) The relay K6 can be used for oscillatory movement.

The voltage is switched over at output 2a from - 9 V to + 9 V via the relay contact K6.

When the output 2a is connected with one of the signal value inputs, the direction can be changed by selecting the corresponding relay and the relay K6 (contact 4c).

c) The ramp generator is bypassed, i.e. ineffective, by selecting the relay d5. As a result, the minimum ramp time of approx. 50 ms is effective.

#### Test Points on Proportional Amplifier:

##### Caution

Measured on DC voltage setting.

1) Measure the supply voltage of + 24 V at the terminals 32ac referred to 26ac.

2) Measure the stabilized voltage  $\pm 9$  V

+ 9 V at 20c with respect to 20a

- 9 V at 26ac with respect to 20a

3) Measure the relay selection voltage (smoothed supply voltage) at 28c with respect to 26ac

4) Measure the signal value voltage of

0 to  $\pm 6$  V at the signal value socket BU1

0 to + 6 V for solenoid A

0 to - 6 V for solenoid B

5) Measure the solenoid currents at the test socket BU3

(solenoid current A) and at BU2 (solenoid current B)

The voltage drop is measured over a resistance of 1 Ohm, i.e. a voltage of 1 V corresponds to 1 A.

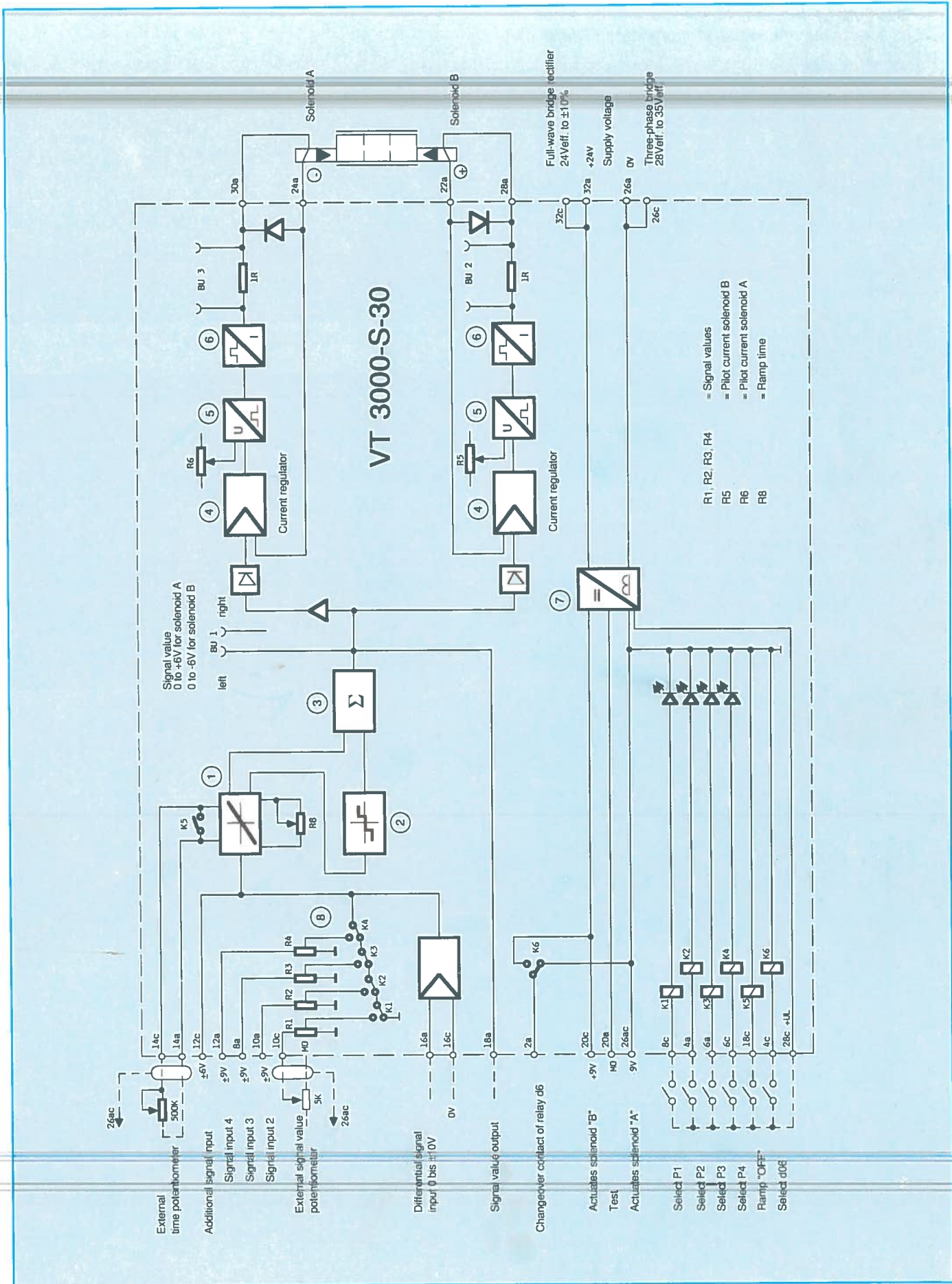


Fig. 19 Terminal connections proportional amplifier Type VT 3000 S 30

The proportional amplifier with 5 variable ramp times serves as a supplement to the proportional amplifier described.

Basically, it is similar to the amplifier with one adjustable ramp time and can also be used for the same applications.

An auxiliary pc board has been added to this amplifier card. This makes it possible to assign an independently variable ramp time to each signal value selection.

Signal value selection R1 is assigned to the ramp time t1 (adjustable at R11)

Signal value selection R2 is assigned to the ramp time t2 (adjustable at R12)

Signal value selection R3 is assigned to the ramp time t3 (adjustable at R13)

Signal value selection R4 is assigned to the ramp time t4 (adjustable at R14)

If all signal values have been selected, the ramp time t5 is effective, adjustable at R10.

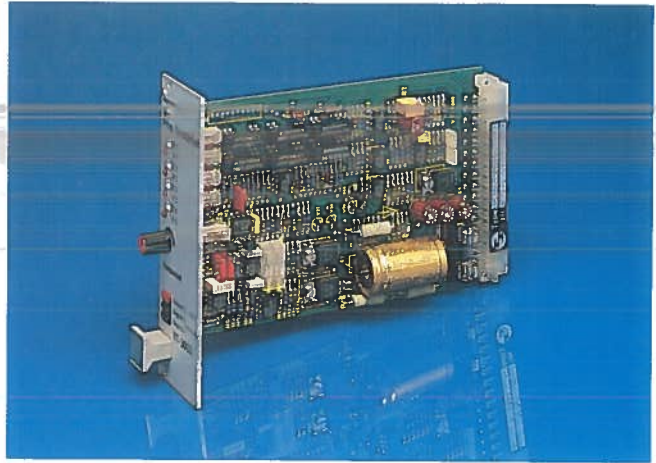


Fig. 20 Proportional amplifier Type VT 3000 S 30

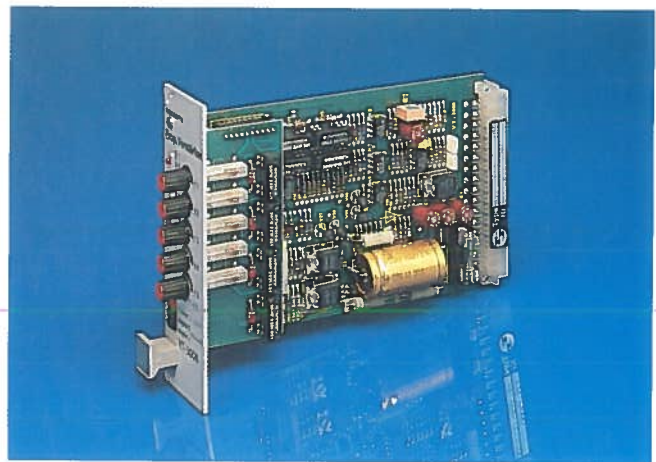


Fig. 21 Proportional amplifier Type VT 3006 S 30

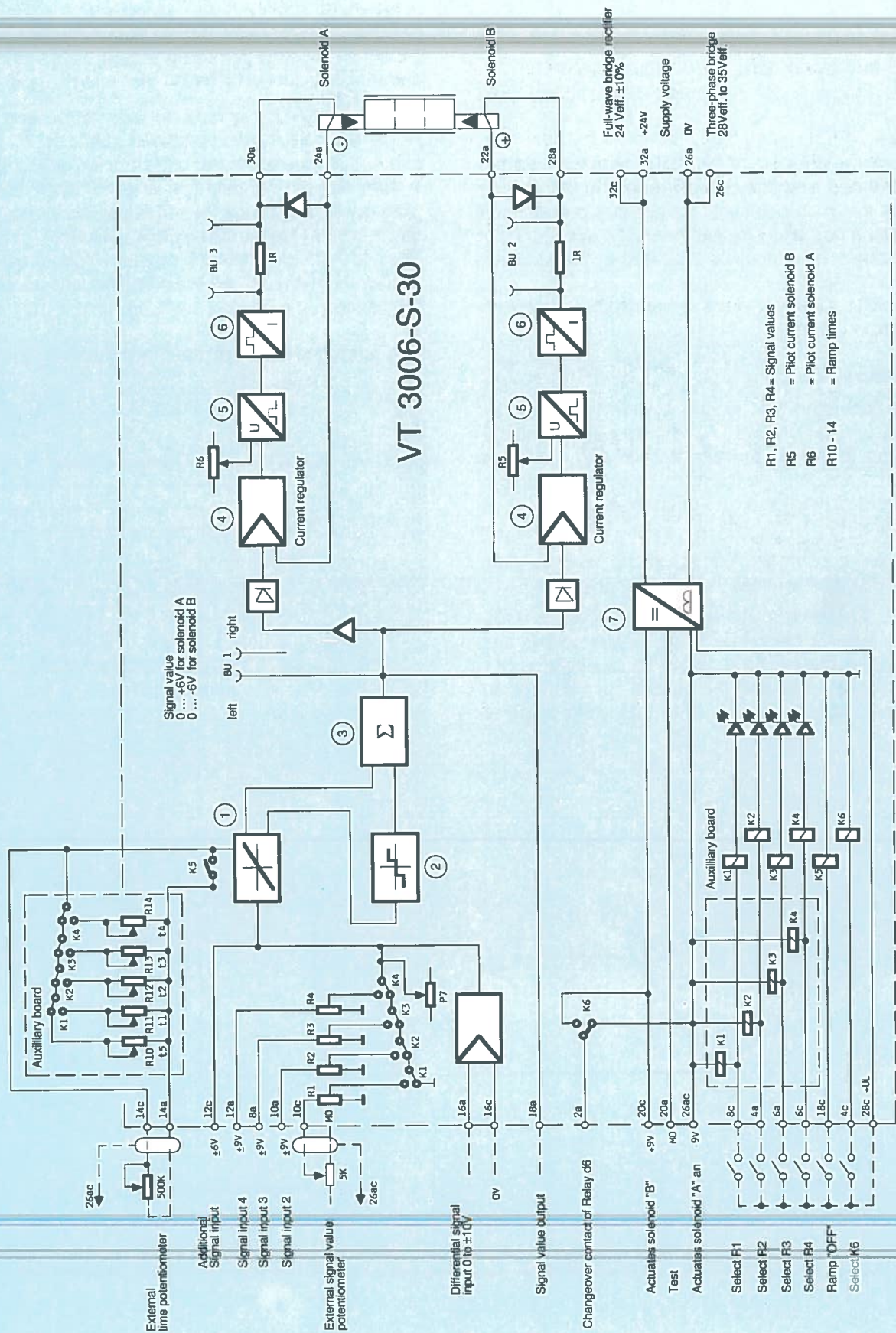


Fig. 22 Terminal connection, proportional amplifier Type VT 3006 S 30

## Control Examples

The following terminal connections remain the same for all controls of the amplifier.

Connection of the solenoid A to 24a and 30a, of the solenoid B to 28a and 22a

- Supply voltage of +24 V between 32ac (+) and 26ac (0 V)

1.) How can a cylinder (or hydraulic motor) be started and accelerated smoothly, decelerated smoothly and stopped at a certain point with the aid of a proportional valve and a proportional amplifier?

The movement sequence is indicated in the velocity/time diagram (Fig. 23).

The amplifier must be wired in accordance with the wiring diagram (Fig. 24).

## Circuit Description

The start command for "extend cylinder" is given by the normally open contact (1). The relays K1 and K2 energize so that only the signal from R2 via K2 is effective as a result of the contacts being connected in series. The rapid traverse rate must therefore be set at potentiometer R2.

The cylinder accelerates corresponding to the ramp time set at R8 until it reaches the velocity set at R2.

When the limit switch (2) is reached, the normally closed contact (2) disconnects the supply to K2 and the relay deenergizes. As a result, R1 is effective (K1 remains switched on) and the cylinder decelerates to creep speed. The relay K1 then also deenergizes

through limit switch(3) and the cylinder slows down to a stop.

The return stroke of the cylinder is initiated by the normally open contact (4), the rapid traverse rate must be set at R4 and the creep speed at R3. The further sequence for cylinder return is analogous to cylinder extension.

Particular attention should be paid to the sequence in which the relays are energised (bearing in mind the sequential arrangement of the internal contacts) to ensure smooth movements, and no "stop" points between switching sequences. If this is not done, intermittent movement of the driven unit will occur.

The acceleration and deceleration values are the same in this example for all acceleration and deceleration procedures.

This ramp time is set with potentiometer R8.

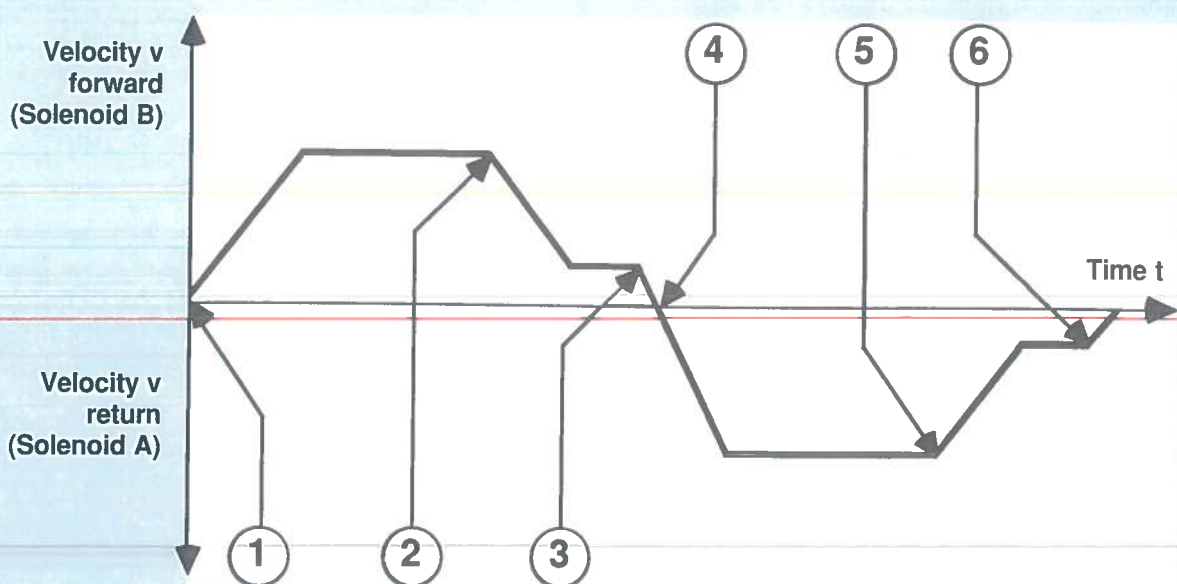
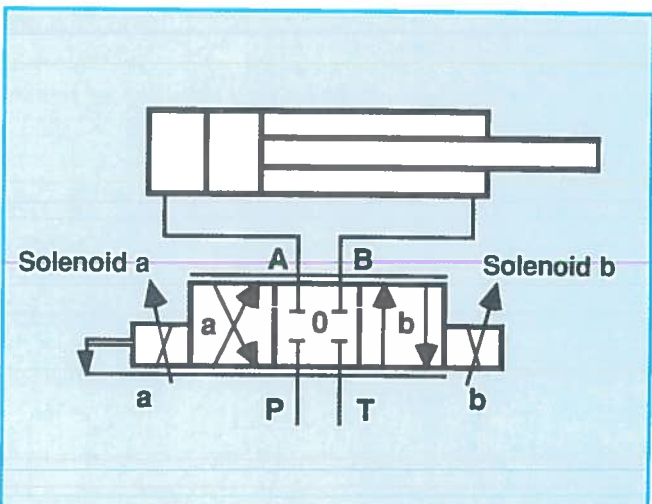


Fig. 23 Velocity/time diagram

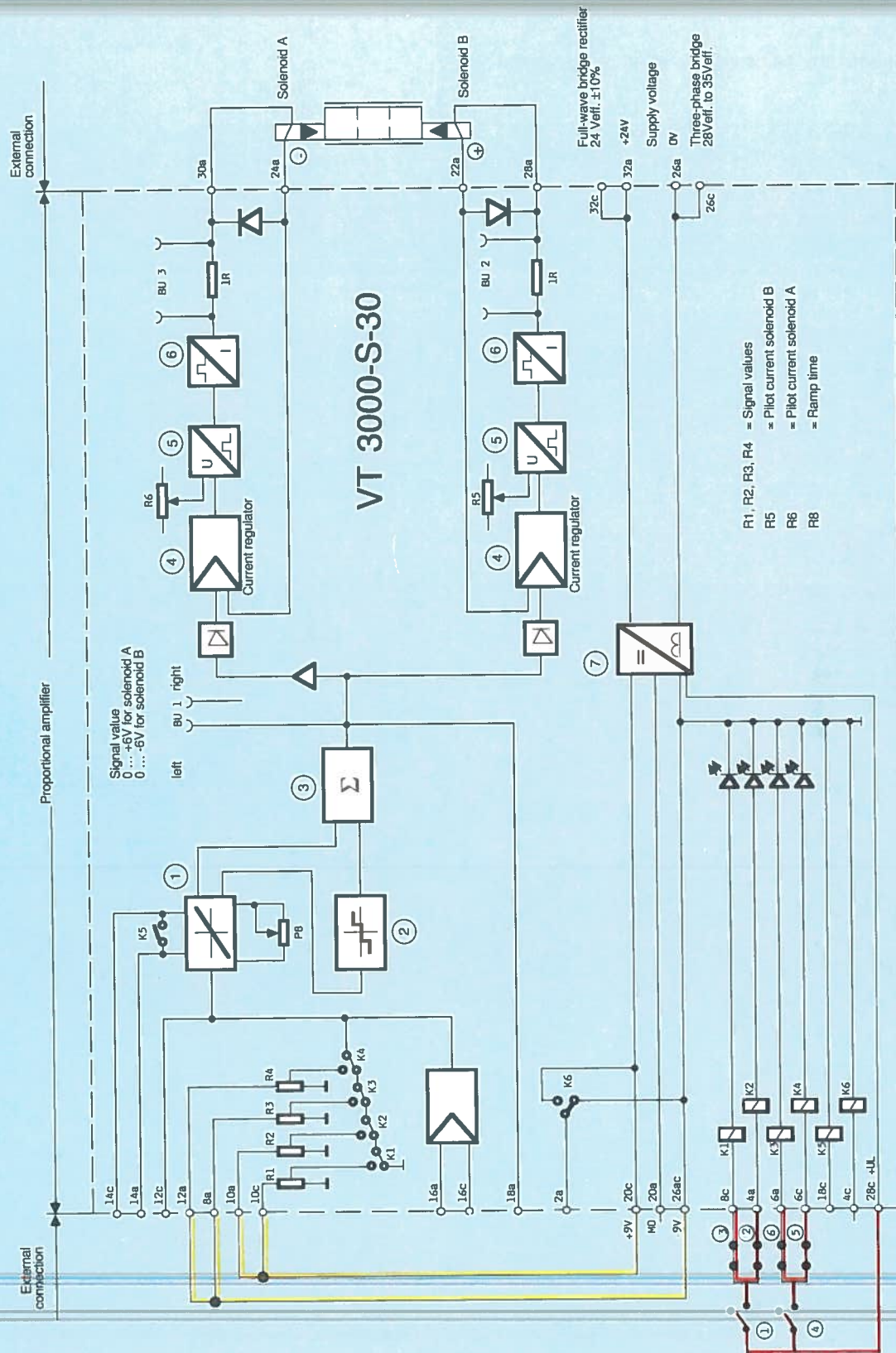


Fig. 24 Wiring diagram

2.) The application is the same as that described in Example 1 but

- the signal value is set via external potentiometers, i.e.

the internal potentiometers R1 to R4 have a limiting function

- the signal values are selected via a stored-program control

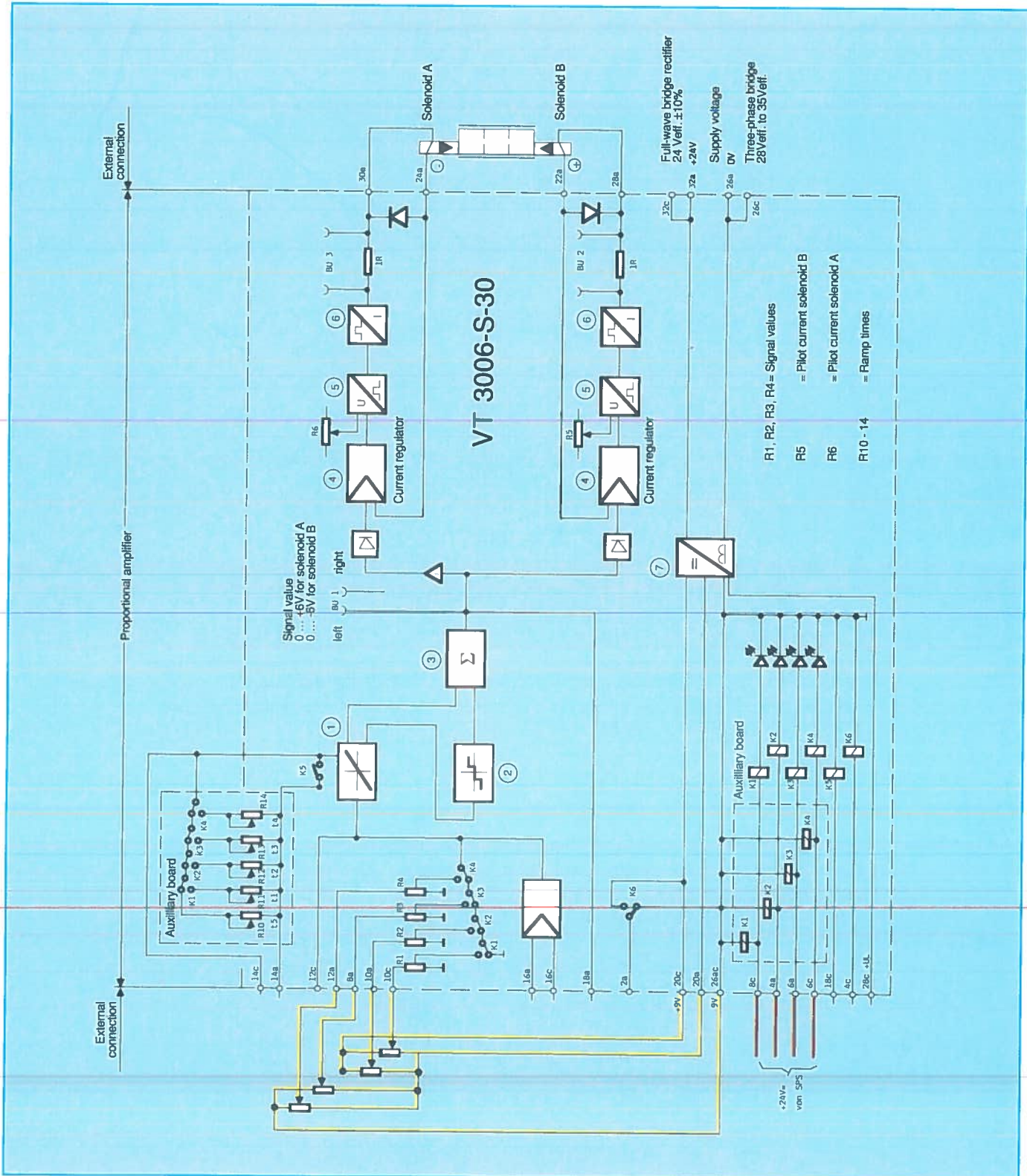


Fig. 25 Wiring diagram

3.) In this case, the two solenoids A and B are energized with an external potentiometer. The external potentiometer receives a stabilized voltage supply of  $\pm 9$  V at both its ends. The external potentiometer is tapped via the input 12a. The internal potentiometer R4 has a limiting function for the external potentiometer.

#### Example

At 100 % signal input at 12a set by the external potentiometer, the value can be changed from 0 to 100 % by means of the internal potentiometer R4.

The set signal value is applied by selecting the relay K4, i.e. solenoid A or solenoid B is active.

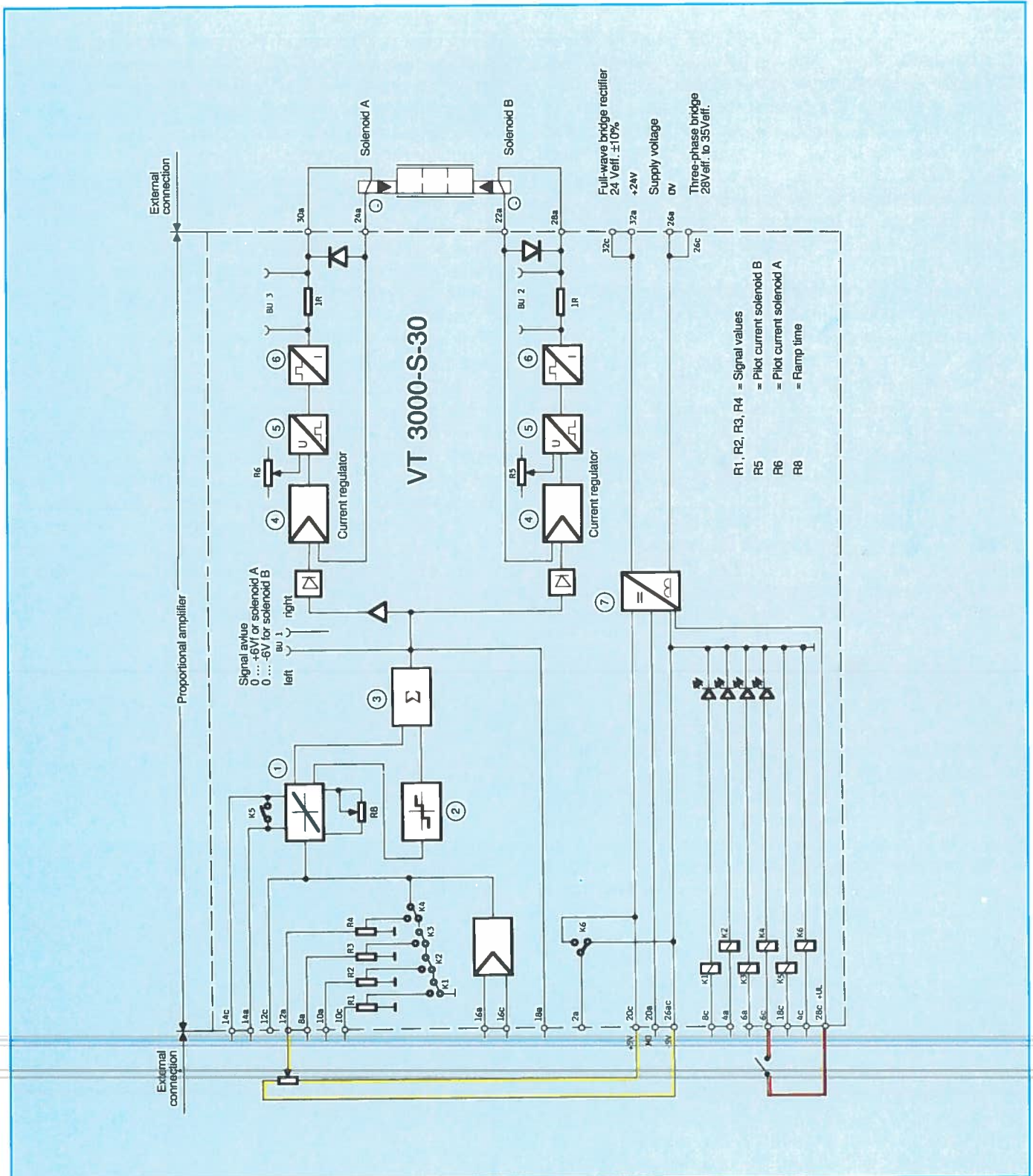


Fig. 26 Wiring diagram

### Proportional Amplifier with Electrical Feedback

#### Proportional Amplifier for Directly Operated Proportional Directional Valves with Feedback

The function of the proportional amplifier is described based on the given block diagram.

The supply voltage for the proportional amplifier board is generated from the consumer network of 220V/380V via transformers with rectifier.

The supply voltage is applied at the terminals 22ac (+) and 28ac (0 V). This supply voltage is smoothed on the amplifier board (9) and is used to produce a stabilized voltage of  $\pm 9$  V.

The stabilized voltage of  $\pm 9$  V is used

a) for the supply of the external potentiometers or the internal potentiometers, available at 26a + 9 V and at 24a - 9 V.

b) for the supply of the internal operational amplifiers.

The amplifier board is equipped with 4 potentiometers for signal value setting R1 to R4 (13).

In order to set a signal value voltage, the 4 signal inputs, terminals 20c, 20a, 14a, 14c must be connected to the stabilized voltage + 9 V terminal 6a or - 9 V terminal 24a.

The solenoid A is active if the signal inputs are connected to + 9 V supply. The solenoid A is connected to the terminals 2a and 32a.

The solenoid B is active if the signal inputs are connected to - 9 V supply. The solenoid B is connected to the terminals 2c and 32c.

The set signal value voltages R1 ... R4 are selected via the relays (12) K1 ... K4. They are applied to terminals 12c, 12a, 16a, 16c.

The selection voltage for the relays can be tapped off at terminal 24c and routed via potential-free contacts to the relay inputs 12c, 12a, 16a, 16c.

A stepped signal is generated at the input of the ramp generator (1) when the set value potentiometers R1 to R4 are selected.

The ramp generator (1) generates a gradually increasing (or decreasing) output signal from a stepped input signal. The rise time (gradient) of the output signal can be adjusted with the potentiometer P5 (ramp time).

The specified ramp time of max. 5 sec. can only be reached over the complete voltage range (from 0 V to  $\pm 6$  V, measured at the signal value test sockets). A signal value voltage of  $\pm 9$  V at the input produces a voltage of  $\pm 6$  V at the signal value test points.

The ramp time is correspondingly shortened if a smaller signal value than  $\pm 9$  V is connected to the input of the ramp generator (1).

The output signal of the ramp generator (1) is routed to the summator (3) and the step function generator (2). The step function generator (2) generates at its output a step function which is added in the summator (3) to the output signal of the ramp generator (1). The step

function is required to move through the zero overlap of the valve quickly.

This step is effective in the case of low signal value voltages (less than 100 mV). The step function generator (2) produces a constant signal if the signal value voltage increases to a higher value.

The output signal of the summator is fed to the PID regulator (4) in the form of a signal value

The oscillator (6) converts a DC signal into an AC voltage (frequency 2.5 kHz). This signal acts on the inductive positional transducer (11).

The positional transducer (11) varies the AC voltage depending on the position of the valve spool. The AC signal is converted back to a DC signal by the demodulator (7).

The matching amplifier (8) amplifies the DC voltage to a maximum voltage of  $\pm 6$  V (max. spool stroke). The output signal of the matching amplifier (8) is fed to the PID regulator (4) as an actual value.

The PID regulator (4) is specially adapted to the type of valve. It produces a signal dependent on the difference between the signal value and actual value. This output signal controls the output stage (5) of the amplifier.

The cable break detection facility (10) monitors the supply line to the positional transducer (11) and cuts the supply to both solenoids (A and B) in the case of fault.

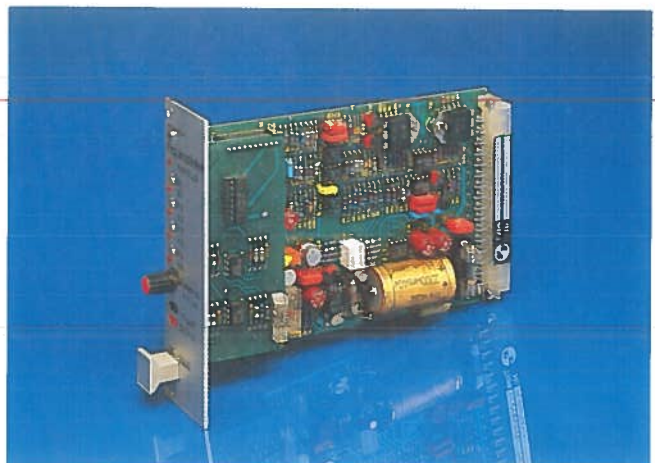


Fig. 27 Proportional amplifier Type VT 5005 S 10

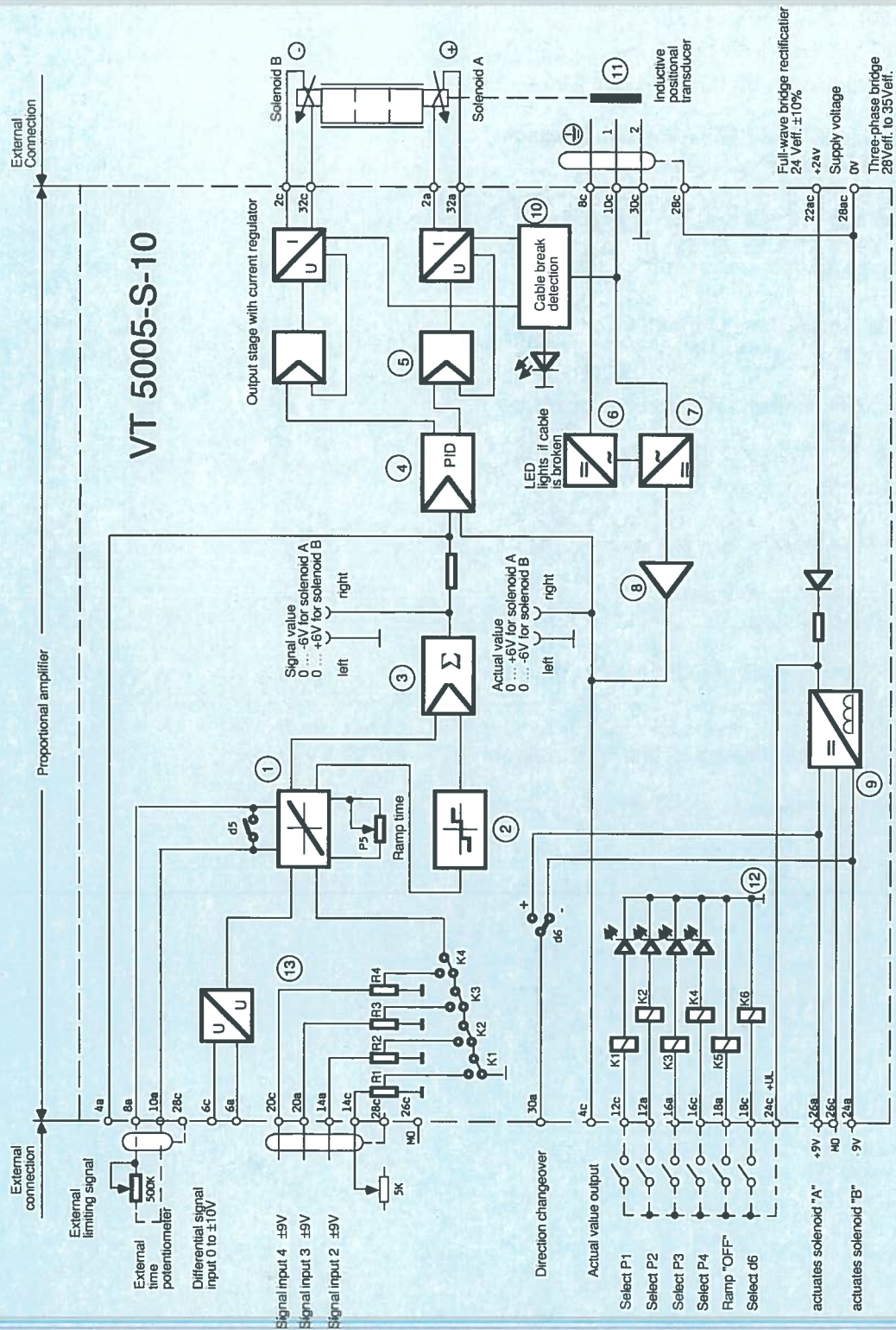


Fig. 28 Terminal connections, proportional amplifier Type VT 5005 S 10

### Additional Notes

a) Differential signal input from 0 to  $\pm 10$  V at 6c and 6a.  
This input is used to achieve high impedance isolation between the valve amplifier card and the external electronic controls.

b) The output 30a is changed over from -9 V to +9 V by selecting the relay k6.  
This results in a polarity reversal of the signal values when the potentiometers are connected to 30a.

c) The ramp generator is bypassed, i.e. ineffective, by selecting the relay k5. As a result, the minimum ramp time of 50 ms is effective.

### Test Points on the Proportional Amplifier

#### Caution

1) Measure the supply voltage +24 V at the terminals 22ac with respect to 28ac

2) Measure the stabilized voltage of  $\pm 9$  V  
+9 V at 26a with respect to 26c  
-9 V at 24a with respect to 26c

3) Measure the relay selection voltage  
+UL at 24c with respect to 28ac

4) Measure the signal value voltage of 0 to  $\pm 6$  V at the signal value test socket

5) Measure the actual value voltage of 0 to  $\pm 6$  V at the actual value test socket  
The actual value corresponds to the spool stroke.

Notes

## Notes

Chapter E

# **Design Criteria for Open Loop Control with Proportional Valves**

Roland Ewald



### Preface

It is necessary to clearly define certain terms relating to calculations for the design of hydraulic control systems. These include definitions such as force directions, velocities, symbols etc. to facilitate calculation by means of computer programs and to improve understanding.

Definitions for cylinder and motor calculation are explained in the following.

## Cylinder Drives

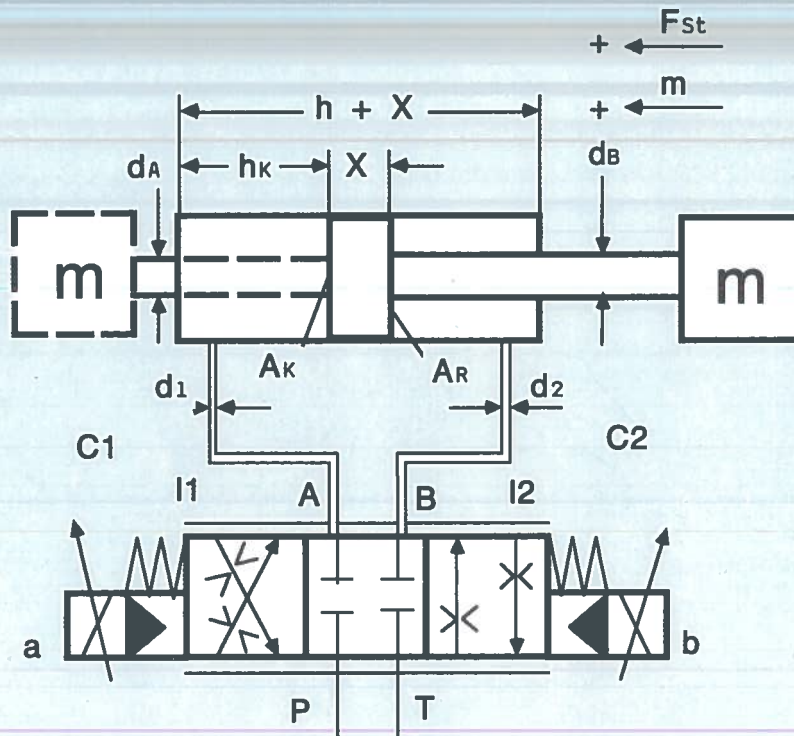


Fig. 1

## Formula Symbols and Dimensions

$D_K$	=	Piston diameter	[mm]	$v$	=	Cylinder - velocity	[m/s]
$d_A$	=	Rod diameter 1, side A	[mm]	$v_A$	=	Oil velocity in pipe, side A	[m/s]
$d_B$	=	Rod diameter 2, side B	[mm]	$v_B$	=	Oil velocity in pipe, side B	[m/s]
$d_1$	=	Pipe diameter, side A	[mm]	$m$	=	Mass moved at cylinder	[kg]
$d_2$	=	Pipe diameter, side B	[mm]	$a$	=	Acceleration	[m/s <sup>2</sup> ]
$h$	=	Cylinder stroke	[mm]	$F_{St}$	=	Static load (from the percentage of mass)	[N]
$s$	=	Travel distance	[mm]	$F_K$	=	Static force (processing or pressing force)	[N]
$h_K$	=	Piston position at min. natural frequency	[mm]	$F_r$	=	Frictional force	[N]
$A_K$	=	Piston area on side A or B	[cm <sup>2</sup> ]	$F_a$	=	Acceleration force	[N]
$A_R$	=	Annulus area on side A or B	[cm <sup>2</sup> ]	$F_G$	=	Total force	[N]
$A_A$	=	Annulus area, side A	[cm <sup>2</sup> ]	$p_P$	=	Pump pressure	[daN/cm <sup>2</sup> ]
$A_B$	=	Annulus area, side B	[cm <sup>2</sup> ]	$\Delta p_v$	=	Pressure losses in pipe	[daN/cm <sup>2</sup> ]
$K_A$	=	Area ratio $A_K/A_A$		$p_a$	=	Acceleration pressure $F_a/(10 \cdot A_W)$	[daN/cm <sup>2</sup> ]
$K_B$	=	Area ratio $A_K/A_B$		$p_D$	=	Dynamic pressure $p_a + (F_s + F_r)/(10 \cdot A_W)$	[daN/cm <sup>2</sup> ]
$A_W$	=	Effective area	[cm <sup>2</sup> ]	$p_s$	=	Static pressure $F_G/(10 \cdot A_W)$	[daN/cm <sup>2</sup> ]
$V_A$	=	Cylinder volume at piston position with minimum natural frequency for annulus area $A_A$ , side A	[cm <sup>3</sup> ]	$\Delta p_1$	=	Pressure drop at control land P→A or A→T	[daN/cm <sup>2</sup> ]
$V_B$	=	Cylinder volume at piston position with minimum natural frequency for annulus area $A_B$ , side B	[cm <sup>3</sup> ]	$\Delta p_2$	=	Pressure drop at control land P→B or B→T	[daN/cm <sup>2</sup> ]
$l_1$	=	Pipe length, side A	[mm]	$p_v$	=	Total pressure drop at valve	[daN/cm <sup>2</sup> ]
$l_2$	=	Pipe length, side B	[mm]	$Q_K$	=	Oil flow for piston area $A_K$	[dm <sup>3</sup> /min]
$V_{L1}$	=	Pipe volume, side A	[cm <sup>3</sup> ]	$Q_R$	=	Oil flow for annulus area $A_R$	[dm <sup>3</sup> /min]
$V_{L2}$	=	Pipe volume, side B	[cm <sup>3</sup> ]	$Q_A$	=	Oil flow for annulus area $A_A$	[dm <sup>3</sup> /min]
$V_3$	=	Total volume, side A	[cm <sup>3</sup> ]	$Q_B$	=	Oil flow for annulus area $A_B$	[dm <sup>3</sup> /min]
$V_4$	=	Total volume, side B	[cm <sup>3</sup> ]				

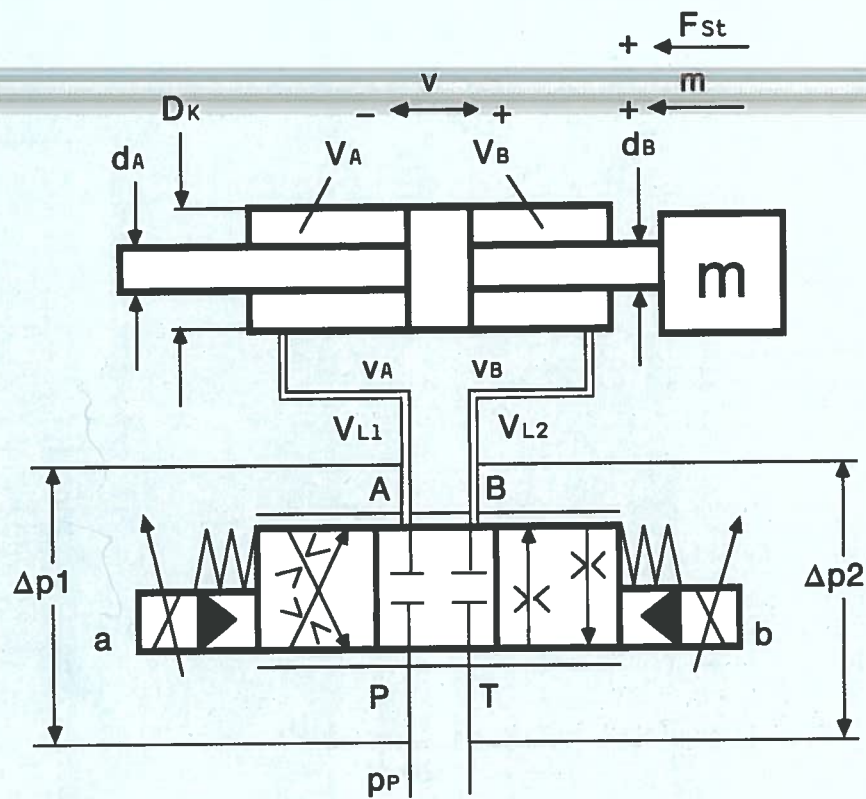


Fig. 2

$Q_P$	= Oil flow at pump connection of proportional valve	[dm <sup>3</sup> /min]
$E_{Oil}$	= Modulus of elasticity of the oil	[kg/cm <sup>2</sup> • sec <sup>2</sup> ]
		$= 1.4 \cdot 10^7$
$C_1$	= Spring constant, side A	[N/m]
$C_2$	= Spring constant, side B	[N/m]
$\omega_0$	= Undamped natural frequency of the system	[1/s]
Hz	= Undamped natural frequency of the system in Hertz	[Hz]
$f_{valve}$	= Critical frequency of the valve in Hertz (Corner frequency at 90° phase lag)	[Hz]
$\omega_V$	= Critical frequency of the valve in (Rad/s) (Corner frequency at 90° phase lag)	[1/s]
$V$	= Overall gain	[1/s]
$\Delta s_V$	= Following error	[mm]
$\Delta s_P$	= Positional error	[mm]
$s_B$	= Acceleration stroke	[mm]
$s_V$	= Distance for constant velocity	[mm]
$s_S$	= Creep speed stroke	[mm]
$v_S$	= Creep speed	[mm/s]
$t_B$	= Acceleration time	[s]
$t_V$	= Time for distance at constant velocity	[s]
$t_S$	= Time for creep speed stroke	[s]
$t_G$	= Overall travel time	[s]

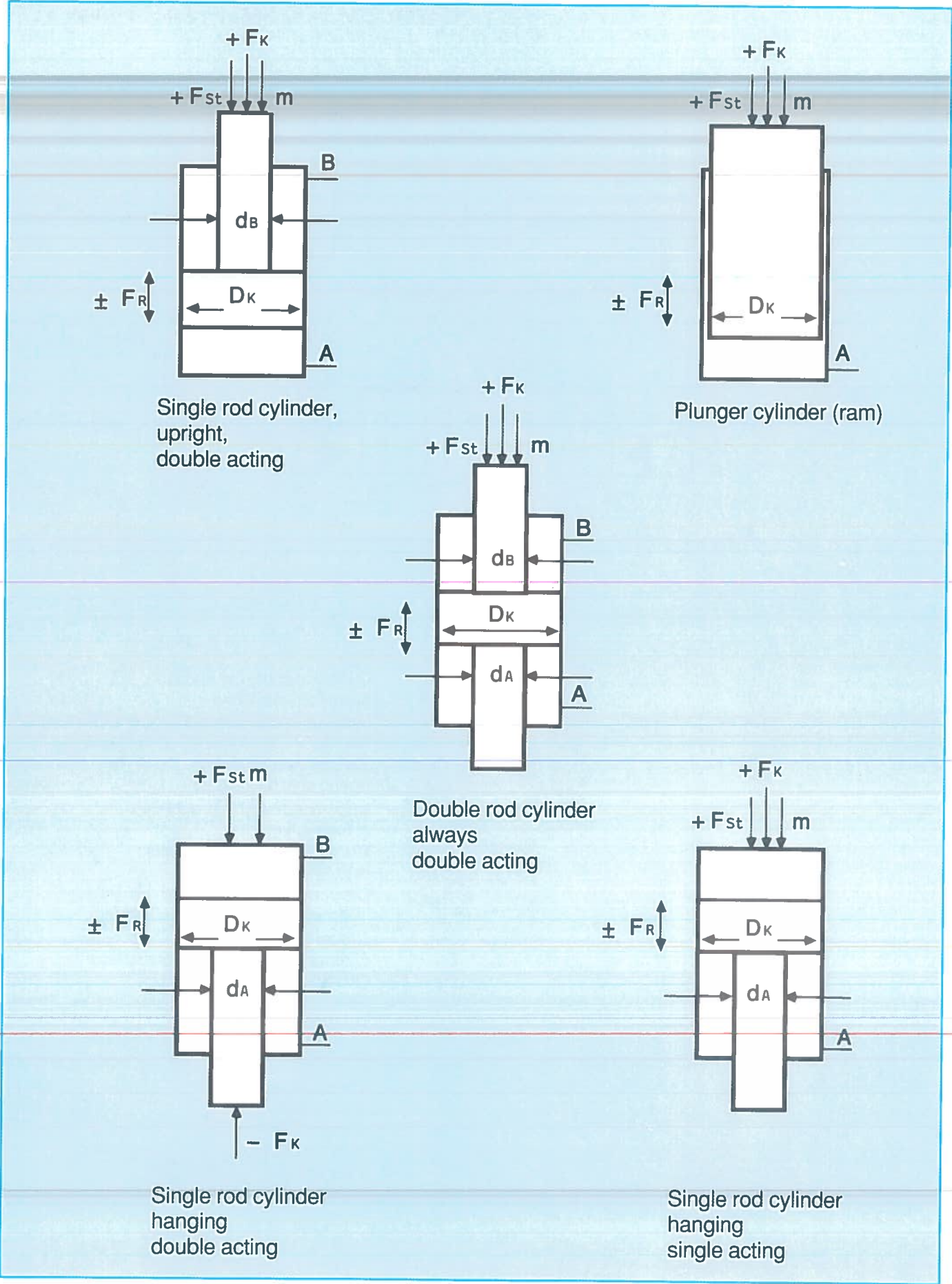


Fig. 3

### Explanation of Masses, Loads, Forces

#### a) Mass $m$

The entire mass moved must be used irrespective of the direction of movement for calculating the accelerating force and the natural frequency.

If there is a step-ratio between the mass moved and the drive, the equivalent (effective) mass must be determined.

The mass changes with the square of a lever or gear ratio

$$m_{\text{equiv}} = m/i^2 \quad [\text{kg}]$$

#### b) Static load $F_{St}$

When lifting and lowering a mass, the mass must be lifted or lowered as a load.

In the case of horizontal movement of a mass, the load is  $F_{St} = 0$ .

The change in load is linear with respect to a lever or gear ratio.

$$F_{St \text{ equiv}} = F_{St} / i \quad [\text{N}]$$

#### c) Static force $F_K$

The drive must produce a static force  $F_K$  to generate a pressing force, forming force or cutting force.

# Motor Drives - Linear and Rotational Output

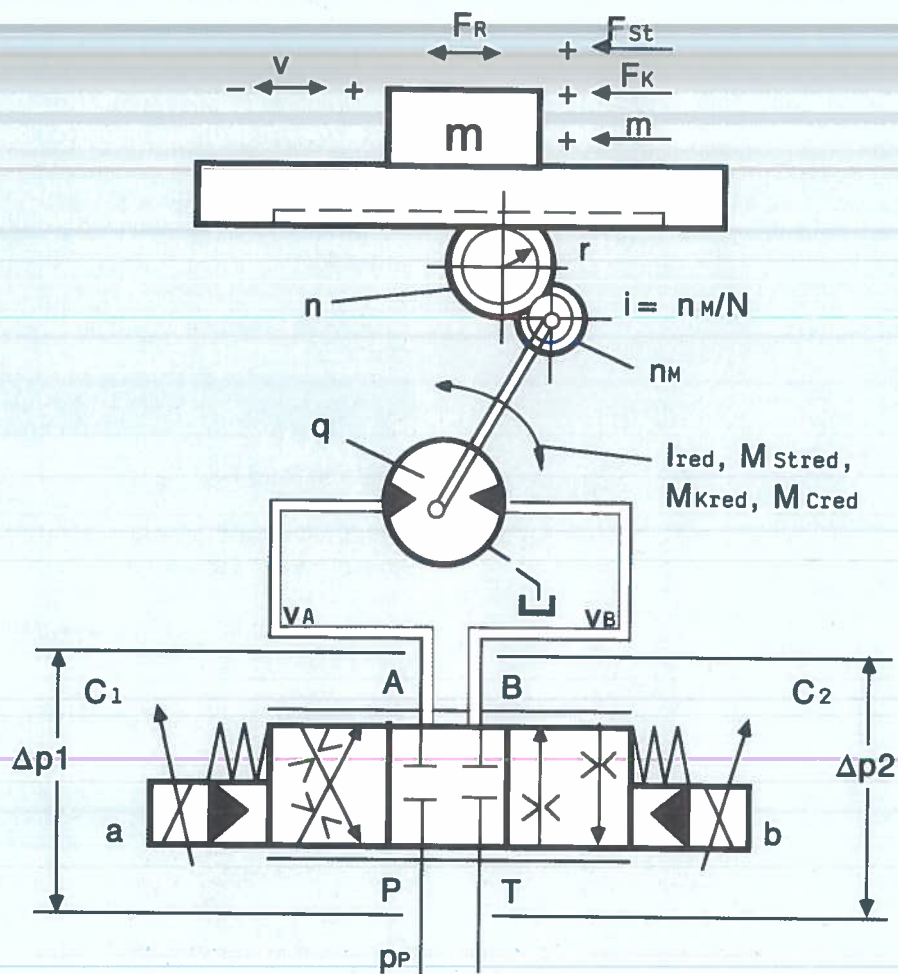


Fig. 4 Motor drives - linear and rotational output

## Formula Symbols and Dimensions

$q$	= Capacity	[cm <sup>3</sup> /U]	$J_M$	= Moment of inertia of motor and gear box	[kgm <sup>2</sup> ]
$d_1$	= Pipe diameter, side A	[mm]	$J_G$	= Overall moment of inertia at output ( $J + J_M \cdot i^2$ )	[kgm <sup>2</sup> ]
$d_2$	= Pipe diameter, side B	[mm]	$J_R$	= Reduced moment of inertia at output ( $J_G/i^2$ )	[kgm <sup>2</sup> ]
$l_1$	= Pipe length, side A	[mm]	$M_{St}$	= Static torque, at output	[Nm]
$l_2$	= Pipe length, side B	[mm]	$M_K$	= Useful torque at output	[Nm]
$V_{L1}$	= Pipe volume, side A	[cm <sup>3</sup> ]	$M_C$	= Frictional force torque at output	[Nm]
$V_{L2}$	= Pipe volume, side B	[cm <sup>3</sup> ]	$p_P$	= Pump pressure	[daN/cm <sup>2</sup> ]
$V_1$	= Overall volume, side A	[cm <sup>3</sup> ]	$\Delta p_v$	= Pressure losses in pipe	[daN/cm <sup>2</sup> ]
$V_2$	= Overall volume, side B	[cm <sup>3</sup> ]	$p_a$	= Acceleration pressure	[daN/cm <sup>2</sup> ]
$v$	= Velocity of load	[m/s]	$p_s$	= Static pressure	[daN/cm <sup>2</sup> ]
$v_A$	= Oil velocity in pipe, side A	[m/s]	$\Delta p_1$	= Pressure drop at control land P→A or A→T	[daN/cm <sup>2</sup> ]
$v_B$	= Oil velocity in pipe, side B	[m/s]	$\Delta p_2$	= Pressure drop at control land P→B or B→T	[daN/cm <sup>2</sup> ]
$\varepsilon$	= Angular acceleration	[1/s <sup>2</sup> ]	$p_{vtotal}$	= Total pressure drop at valve	[daN/cm <sup>2</sup> ]
$n$	= Speed of output shaft	[1/min]	$Q_p$	= Oil flow at pump connection of proportional valve	[L/min]
$n_M$	= Speed of input shaft	[1/min]			
$m$	= Mass moved	[kg]			
$F_{St}$	= Static load	[N]			
$F_K$	= Static force	[N]			
$F_R$	= Frictional force	[N]			
$J$	= Moment of inertia at output	[kgm <sup>2</sup> ]			

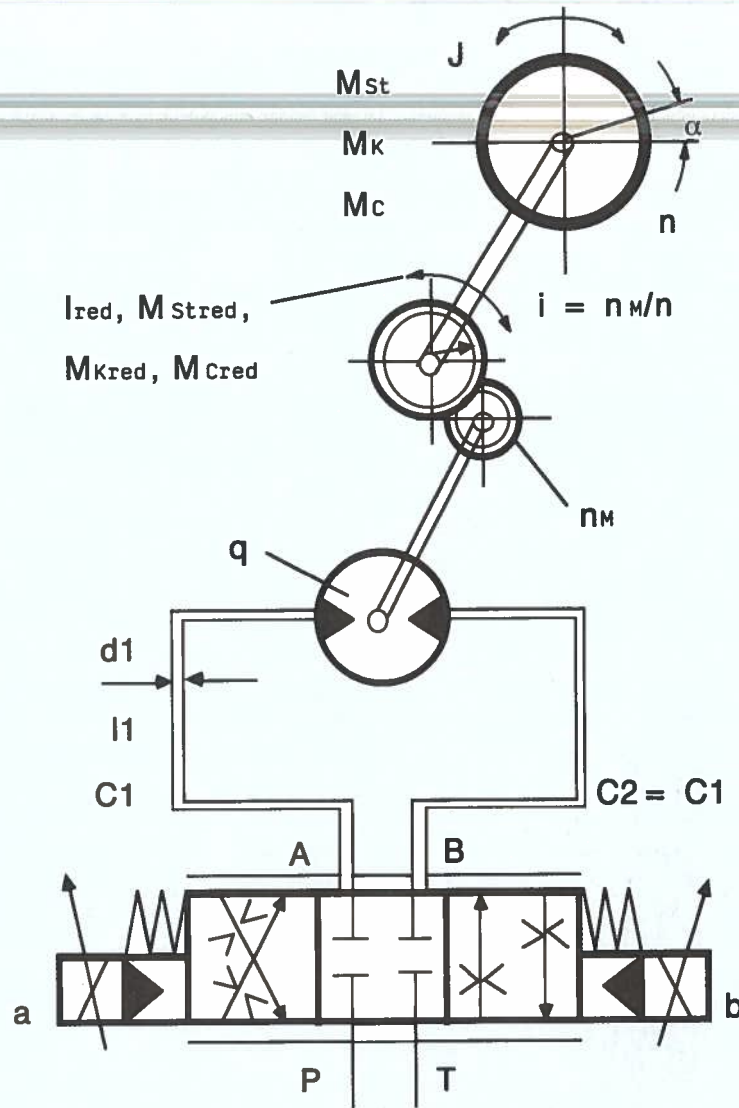


Fig. 5 Motor drive - rotational output

$E_{Oil}$	= Modulus of elasticity of oil		$s$	= Travel distance	[mm]
	$= 1.4 \cdot 10^7$	[kg/cm $\cdot$ sec $^2$ ]	$s_B$	= Acceleration distance	[mm]
$C_1$	= Spring constant, side A	[N/m]	$s_v$	= Distance for constant velocity	[mm]
$C_2$	= Spring constant, side B	[N/m]	$s_s$	= Creep speed stroke	[mm]
$\omega_0$	= Undamped natural frequency of system	[1/s]	$\alpha$	= Travel angle at output	[ $^\circ$ ]
$hz$	= Undamped natural frequency of system in Hertz	[Hz]	$\alpha_B$	= Acceleration angle at output	[ $^\circ$ ]
$f_{valve}$	= Critical frequency of valve in Hertz (corner frequency at 90 $^\circ$ phase lag)	[Hz]	$\alpha_v$	= Travel angle for $n = \text{constant}$ at output	[ $^\circ$ ]
$\omega_v$	= Critical frequency of valve in (Rad/s) (corner frequency-at 90 $^\circ$ phase lag)	[1/s]	$v_s$	= Creep speed (linear)	[mm/s]
$V$	= Overall gain	[1/s]	$n_s$	= Rotational creep speed at output	[1/s]
$\Delta s_v$	= Following error (linear)	[mm]	$t_B$	= Acceleration time	[s]
$\Delta \beta_x$	= Following error at drive motor	[mm]	$t_v$	= Time for travel at constant velocity	[s]
$\Delta \alpha_x$	= Following error at output shaft	[mm]	$t_s$	= Time for creep speed stroke	[s]
$\Delta X_p$	= Positional error (linear)	[mm]	$t_G$	= Overall travel time	[s]
$P_M$	= Positional error at drive motor	[ $^\circ$ ]			
$P_W$	= Positional error at output shaft	[ $^\circ$ ]			

Corresponding to the main application of proportional directional valves, i.e. acceleration, traverse and deceleration of hydraulically moved masses, the required acceleration or deceleration must be determined when designing the open loop control.

However, any arbitrary selection cannot be made.

The possible value for acceleration or deceleration depends on various factors:

### 1) Deceleration and acceleration time for constant acceleration

Fig. 6 shows the physical relationship between acceleration time acceleration and the velocity to be reached.

Acceleration time

$$t_B = \frac{v}{a} \quad [s]$$

$$a = \frac{v}{t_B} \quad [m/sec^2]$$

$v$  = Velocity [m/s]

$a$  = Acceleration [m/sec<sup>2</sup>]

$t_B$  = Acceleration time [s]

The feasible acceleration time for a certain final velocity is clearly shown by the resulting curves.

The acceleration should not be selected too high (lower limit; see Fig. 6 yellow line) since the time gain would only be extremely slight.

An acceleration which is too low (left limit; see Fig. 6 green line) results in an extremely long acceleration time.

The diagram clearly shows that the ramp time adjustable between 0.1 s and 5 s is more than sufficient for most applications.

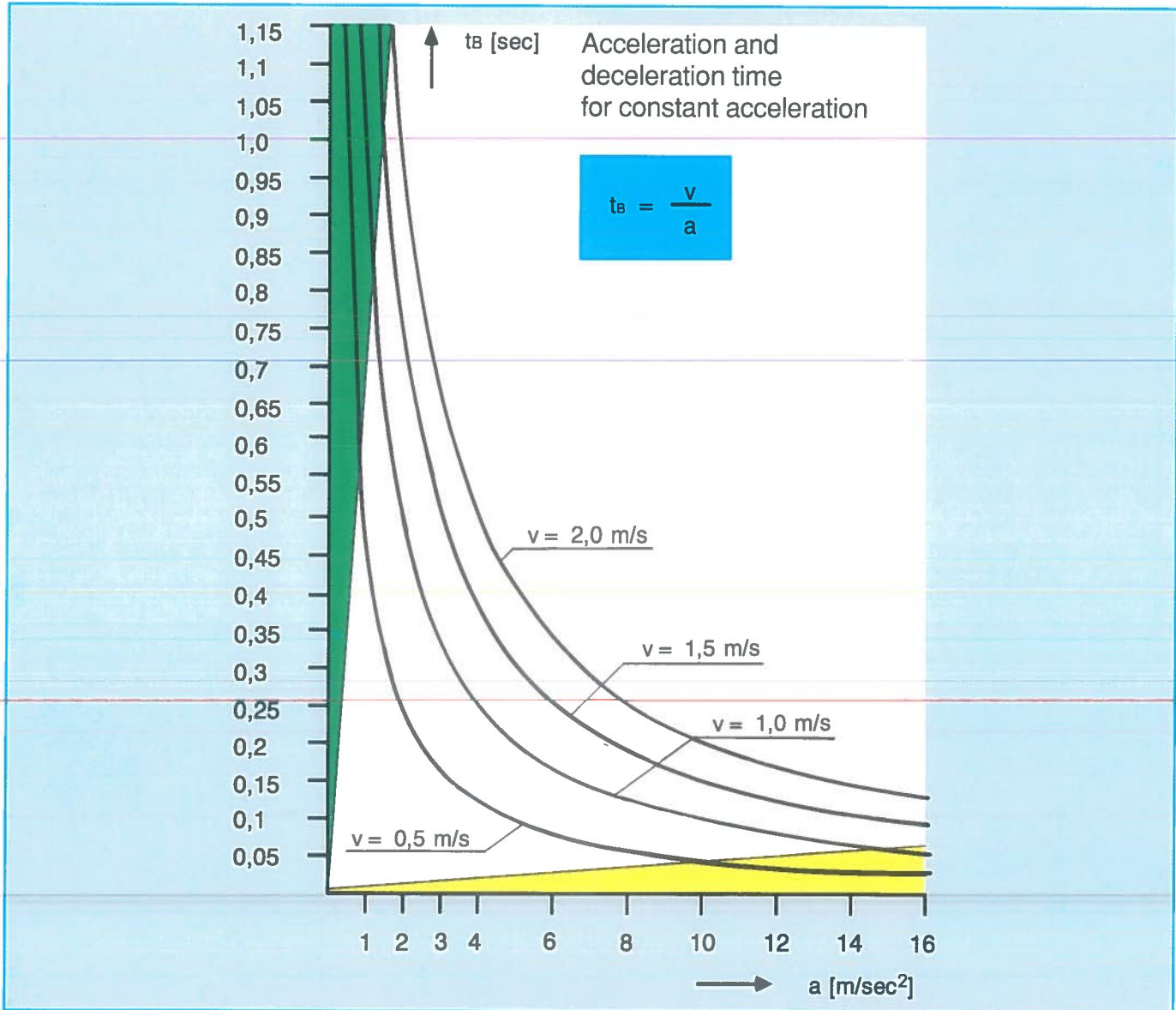


Fig. 6 Acceleration and deceleration time for constant acceleration

## 2) Acceleration and deceleration distance for constant acceleration

Fig. 7 clearly shows the physical relationship between acceleration or deceleration distance, acceleration and velocity.

Acceleration or deceleration distance

$$s_B = (v^2/2a) \cdot 10^3 \quad [\text{mm}]$$

or

$$s_B = 1/2 \cdot a \cdot t_B^2 \cdot 10^3 \quad [\text{mm}]$$

In order to decelerate a mass from one speed to another, a specific distance is required. This is often chosen by "feel", and is quite often simply too short.

In this case, it should be noted that the acceleration or deceleration distance changes as the square of the velocity (see Fig. 7 red dotted lines).

Double travel speed requires 4 times the distance for acceleration or deceleration.

Also referred to distance, it can be clearly seen that an increase in acceleration to excessively high values is not practical (see Fig. 7 yellow area).

In addition, the energy to be installed must also be taken into consideration when selecting the acceleration:

Acceleration force

$$F_a = m \cdot a \quad [\text{N}]$$

Acceleration pressure

$$p_a = F_a/10/A_W \quad [\text{daN/cm}^2]$$

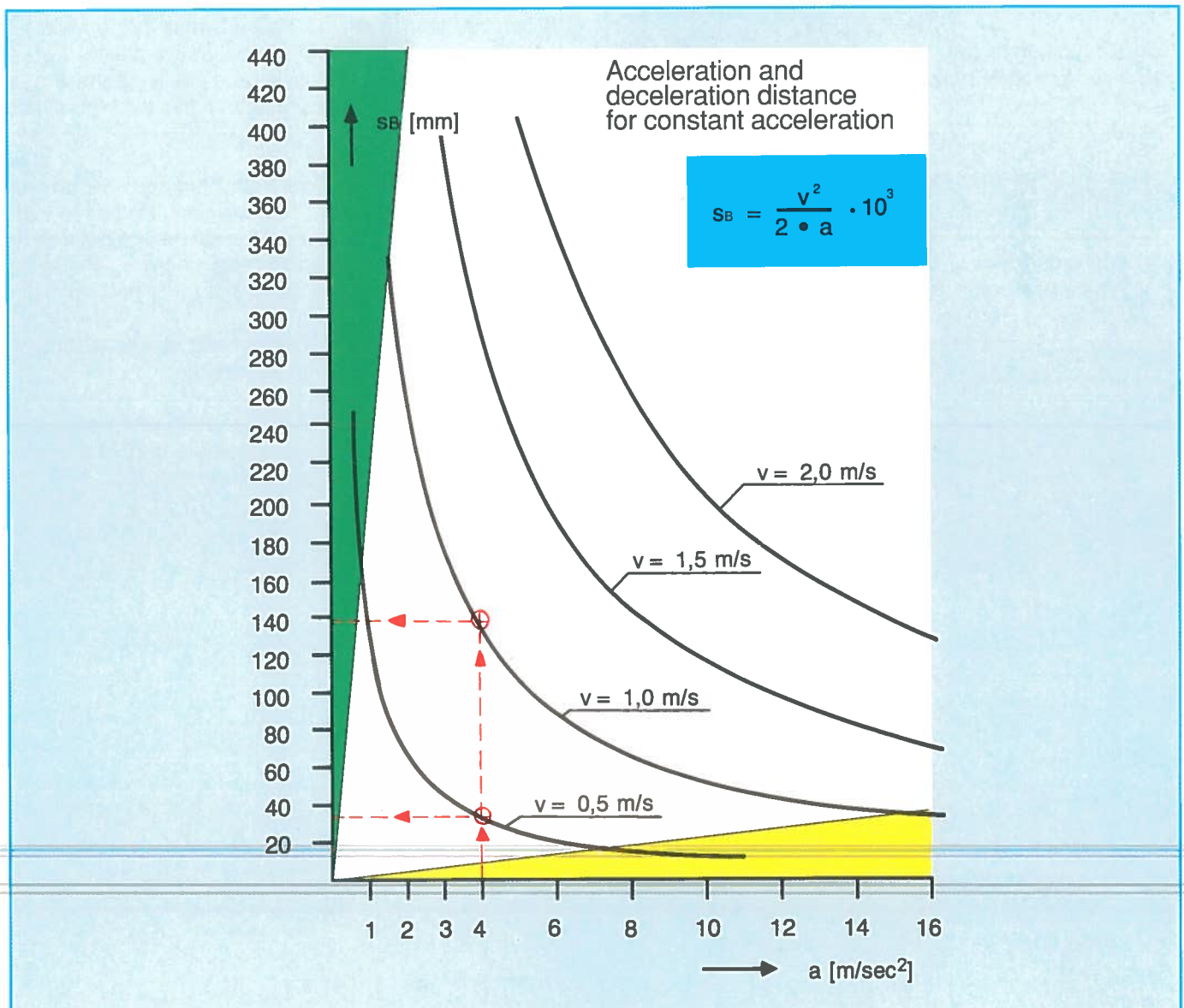


Fig. 7 Acceleration and deceleration distance for constant acceleration.

## 3) Natural Frequency

A further important point for consideration with regard to the selection of the acceleration is the natural frequency. It is a measure for the stability, the stiffness of a system.

If, without taking into consideration the natural frequency, the acceleration is selected too high, or if the natural frequency is too low, then the system will oscillate.

For the driven unit, cylinder or a motor, this means irregular movements.

Similar to a mechanical spring/mass system, the natural frequency of a hydraulically driven unit can be calculated from the spring constant  $C$  and mass  $M$  moved with the formula

$$\omega_0 = \sqrt{C/m} \quad [1/s]$$

$C$  = Spring constant [Nm]  
 $m$  = Mass [kg]

Correspondingly, the natural frequency can be calculated for rotary movement with the formula

$$\omega_0 = \sqrt{C/J} \quad [1/s]$$

$C$  = Spring constant [N/rad]  
 $J$  = Moment of inertia [kgm<sup>2</sup>]

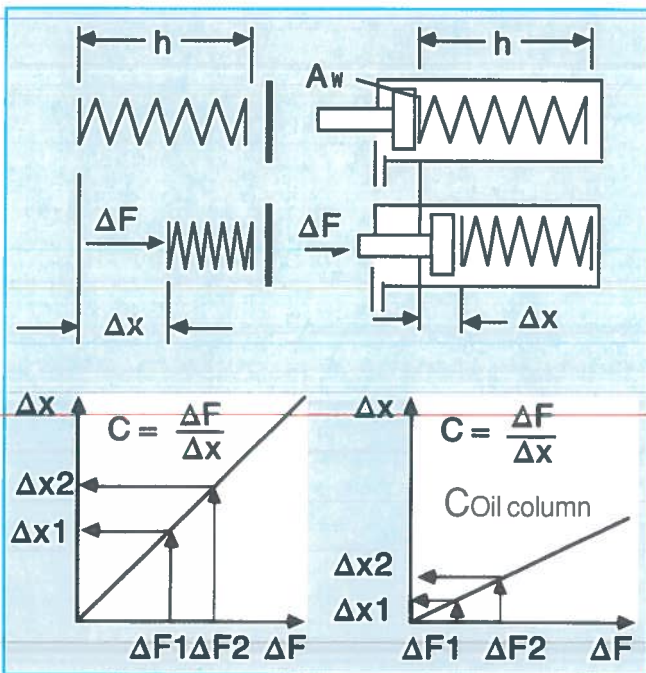


Fig. 8 Comparison mechanical/hydraulic spring

System stiffness:

Comparison mechanical/hydraulic spring for cylinder

$$\begin{aligned} C &= \Delta F / \Delta X \\ &= E_{oil} \cdot A_w / (h/10) \quad [kg/s^2] \\ &= [N/m] \end{aligned}$$

The spring constant for the rotary movement is calculated analogously

$$\begin{aligned} C &= [V_G / (2 \cdot \pi)]^2 \cdot E_{oil} / [(V_G / 2) 10^4] \\ &= V_G \cdot E_{oil} / 2 \cdot \pi^2 \cdot 10^4 \quad [kg \cdot m^2 / s^2 \cdot Rad] \\ &= [Nm/Rad] \end{aligned}$$

Rad = 1

It can be seen from the diagram and the formula for calculating the spring constant that the piston area  $A$  must be as large as possible and the length of the oil column  $h$  as small as possible to achieve a high spring constant  $C$ .

These are the theoretical relationships. In practical applications, however, the working distances and therefore the necessary cylinder stroke are defined by the physical requirements of the machine. The effective piston area  $A_w$  can, however, be varied relatively easily.

The pipes between the cylinder and flow "control unit" should be arranged as short as possible.

The length of pipe between the pump and valve is of no significance provided the pressure cannot collapse when the supply of oil is suddenly interrupted.

### Pressure Ratios at the Throttle Edges of the Valve During the Acceleration and Deceleration Phases, and at Constant Velocity

Various forces are necessary at the cylinder or hydraulic motor for the individual movement phases.

At constant pump pressure, the pressure drop at the control lands of a proportional valve will therefore differ correspondingly.

This is illustrated based on the following example.

Given:

$m$	$= 700$	[kg]
$F$	$= 7000$	[N]
$F_{St}$	$= F \cdot \sin 30^\circ = 7000 \cdot 0.5 = 3500$	[N]
$v$	$= 2.0$	[m/s]
$s_B$	$= 250$	[mm]
$F_R$	$= 0$	[N]

(Frictional force ( $F_R$ ) is not included in this calculation.)

Acceleration

$a$	$= v^2 / (2 \cdot s_B \cdot 10^{-3})$	[m/s <sup>2</sup> ]
$a$	$= 2^2 / (2 \cdot 250 \cdot 10^{-3}) = 8$	[m/s <sup>2</sup> ]

Acceleration time

$t_B$	$= v/a = 2/8 = 0.25$	[s]
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Note:

The acceleration in a throttle control is an average acceleration.

Forces necessary during up-stroke:

$F_{St}$	$= 3500$	[N]
$F_A$	$= m \cdot a = 700 \cdot 8 = 5600$	[N]

Acceleration

$F_G$	$= F_{St} + F_A = 3500 + 5600 = 9100$	[N]
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Constant velocity

$F_G$	$= F_{St} = 3500$	[N]
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Deceleration

$F_G$	$= F_{St} - F_A = 3500 - 5600 = -2100$	[N]
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Forces necessary during down-stroke:

Acceleration

$F_G$	$= -F_{St} + F_A = -3500 + 5600 = 2100$	[N]
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Constant velocity

$F_G$	$= -F_{St} = -3500$	[N]
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Deceleration

$F_G$	$= -F_{St} - F_A = -3500 - 5600 = -9100$	[N]
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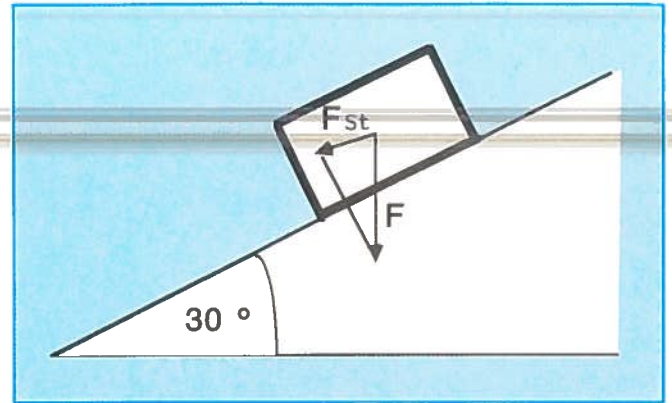


Fig. 9

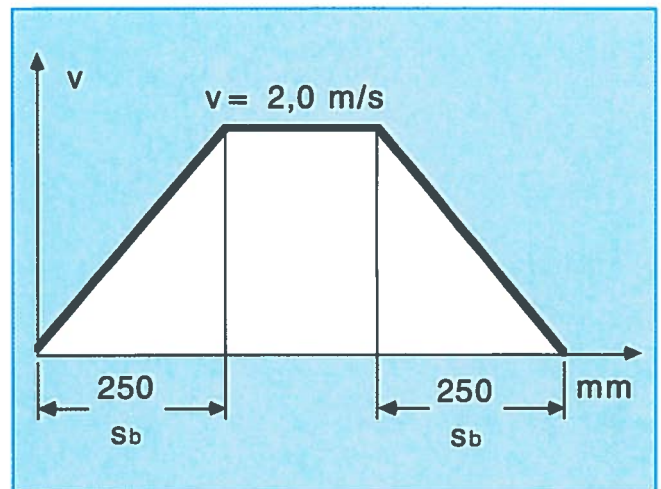


Fig. 10

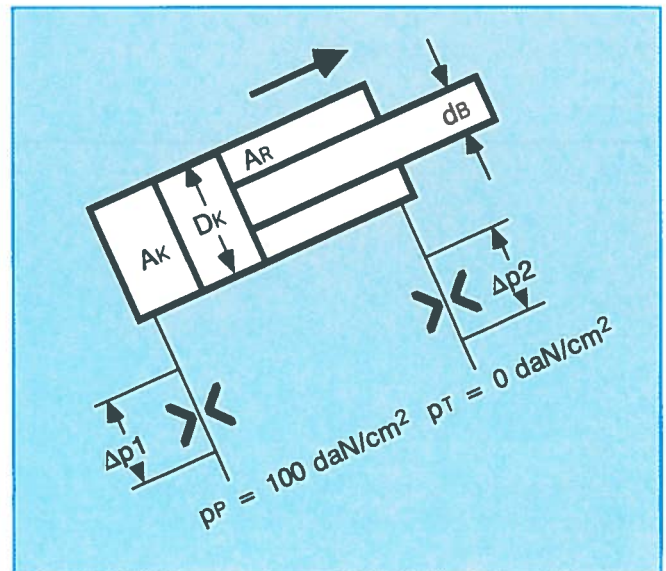


Fig. 11

## Cylinder Dimensions, Oil Flow and System Pressure

$D_K$	= 50	[mm]
$d_B$	= 36	[mm]
$A_K$	= 19.64	[cm <sup>2</sup> ]
$A_R$	= 9.45	[cm <sup>2</sup> ]
$h$	= 700	[mm]
$Q_{\max, AK}$	= 235.6	[dm <sup>3</sup> /min]
$Q_{\max, AR}$	= 113.4	[dm <sup>3</sup> /min]
$p_P$	= 100	[daN/cm <sup>2</sup> ]

The pump pressure of  $p_P = 100$  ([daN/cm<sup>2</sup>]) was decided upon for this applicaton (accumulator system).

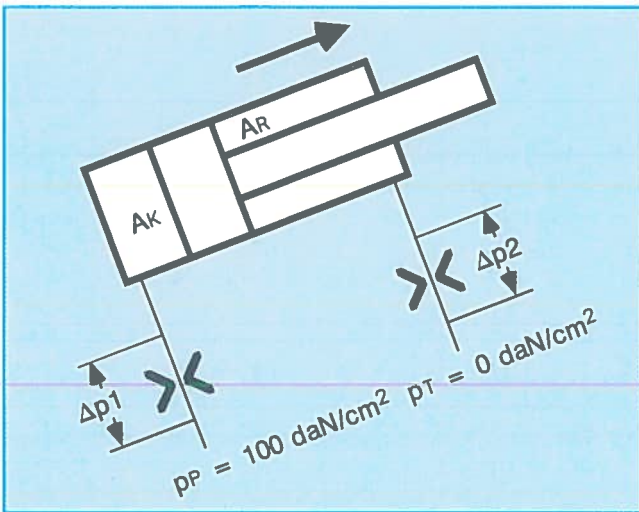


Fig. 12 Up stroke

Which pressures are produced during the individual movement phases?

### a) Up stroke

for  $p_1 = p_2 \rightarrow$

$$F_G/10 = A_K \cdot (p_P - \Delta p_1) - A_R \cdot \Delta p_2$$

$$F_G/10 = A_K \cdot p_P - A_K \cdot \Delta p_1 - A_R \cdot \Delta p_1$$

$$\Delta p_1 = (A_K \cdot p_P - F_G/10) / (A_K + A_R)$$

### Acceleration

$$\Delta p_1 = (19.64 \cdot 100 - 9100/10) / (19.64 + 9.45) \approx 36 \text{ [daN/cm}^2\text{]}$$

$$p_V = 2 \cdot \Delta p_1 \approx 72 \text{ [daN/cm}^2\text{]}$$

### Constant velocity (see Figs. 14, 15 and red line)

$$\Delta p_1 = (19.64 \cdot 100 - 3500/10) / (19.64 + 9.45) \approx 55 \text{ [daN/cm}^2\text{]}$$

$$p_V = 2 \cdot \Delta p_1 \approx 110 \text{ [daN/cm}^2\text{]}$$

### Deceleration (see Figs. 14, 15 and red dotted line)

$$\Delta p_1 = (19.64 \cdot 100 + 2100/10) / (19.64 + 9.45) \approx 75 \text{ [daN/cm}^2\text{]}$$

$$p_V = 2 \cdot \Delta p_1 \approx 150 \text{ [daN/cm}^2\text{]}$$

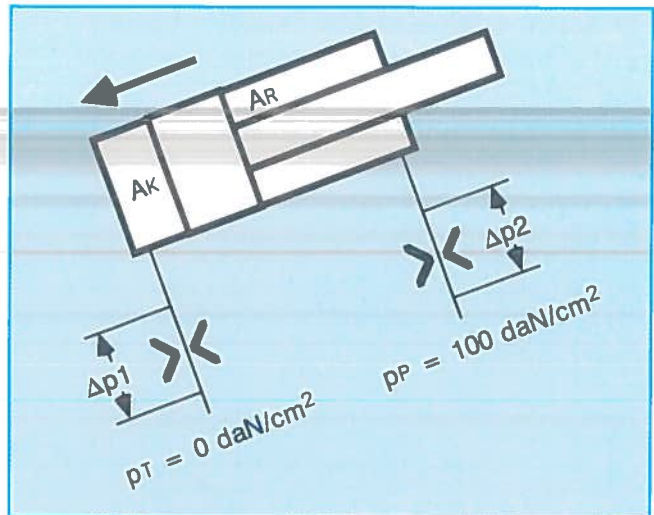


Fig. 13 Down stroke

### b) Down stroke

for  $\Delta p_1 = \Delta p_2 \rightarrow$

$$F_G/10 = A_R \cdot (p_P - \Delta p_2) - A_K \cdot \Delta p_1$$

$$F_G/10 = A_R \cdot p_P - A_R \cdot \Delta p_1 - A_K \cdot \Delta p_1$$

$$p_1 = (A_R \cdot p_P - F_G/10) / (A_K + A_R)$$

### Acceleration

$$\Delta p_1 = (9.45 \cdot 100 - 2100/10) / (19.64 + 9.45) \approx 25 \text{ [daN/cm}^2\text{]}$$

### Constant velocity

$$\Delta p_1 = (9.45 \cdot 100 + 3100/10) / (19.64 + 9.45) \approx 43 \text{ [daN/cm}^2\text{]}$$

$$p_V = 2 \cdot \Delta p_1 \approx 86 \text{ [daN/cm}^2\text{]}$$

### Deceleration

$$\Delta p_1 = (9.45 \cdot 100 + 9100/10) / (19.64 + 9.45) \approx 64 \text{ [daN/cm}^2\text{]}$$

$$p_V = 2 \cdot \Delta p_1 \approx 128 \text{ [daN/cm}^2\text{]}$$

During commissioning of the open loop control, the proportional directional valve

Type 4WRZ16E1-100 ...

( $Q = 100 \text{ dm}^3/\text{min}$  at  $\Delta p = 10 \text{ daN/cm}^2$  with control land ratio 2:1)

has proved to be the most suitable.

Recalculation of the pressure drops at the control lands of the proportional valve and the corresponding percentage openings also confirm this statement.

For the upwards travel (see Figs. 14 and 15 red line), the pressure drops at the control lands and the associated percentage of control current can also be illustrated.

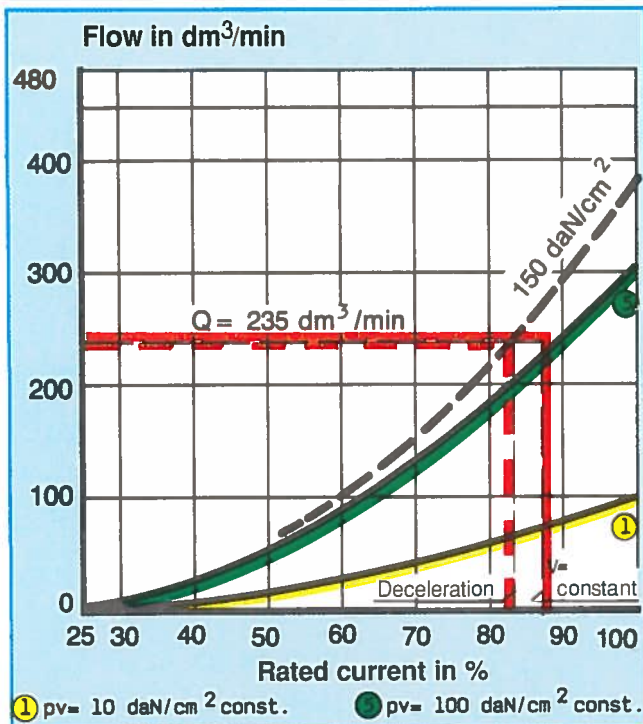
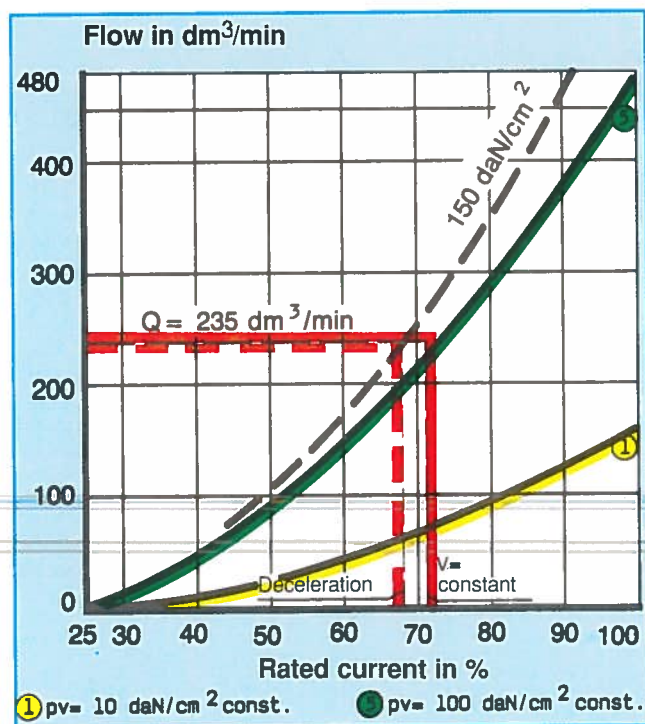


Fig. 14 Flow/current curve for  $100 \text{ dm}^3/\text{min}$  nominal flow at  $10 \text{ daN/cm}^2$  valve pressure drop

The comparison with the proportional valve Type 4WRZ16E1 - 150 ...

( $Q = 150 \text{ dm}^3/\text{min}$  at  $p = 10 \text{ daN/cm}^2$  with control land ration 2:1) shows that the spool is too large, therefore producing poor flow resolution.



Also the reduction of the pump pressure in the interest of higher hydraulic efficiency is not recommended for this control.

The minimum required pressure can be calculated from the maximum total force during acceleration and the minimum total pressure drop ( $\geq 10 \text{ daN/cm}^2$ ) at the control lands of the throttle element.

#### Calculation of the necessary pump pressure during:

##### Up stroke

$$\begin{aligned} \text{for } \Delta p_1 &= \Delta p_2 = 5 \text{ daN/cm}^2 \rightarrow \\ F_G/10 &= A_K \cdot (p_P - \Delta p_1) - A_R \cdot \Delta p_2 \\ F_G/10 &= A_K \cdot p_P - A_K \cdot \Delta p_1 - A_R \cdot \Delta p_1 \\ p_P &= [F_G + \Delta p_1 \cdot (A_K + A_R)] / A_K \\ &= [9100/10 + 5 \cdot (19.64 + 9.45)] / 19.64 \\ &\approx 54 \text{ daN/cm}^2 \end{aligned}$$

##### Down stroke

$$\begin{aligned} \text{for } \Delta p_1 &= \Delta p_2 = 5 \text{ daN/cm}^2 \rightarrow \\ F_G &= A_R \cdot (p_P - \Delta p_2) - A_K \cdot \Delta p_1 \\ F_G &= A_R \cdot p_P - A_R \cdot \Delta p_1 - A_K \cdot \Delta p_1 \\ p_P &= [F_G + \Delta p_1 \cdot (A_K + A_R)] / A_R \\ &= [2100/10 + 5 \cdot (19.64 + 9.45)] / 9.45 \\ &\approx 38 \text{ daN/cm}^2 \end{aligned}$$

Selected pump pressure:  $p_P = 55 \text{ daN/cm}^2$

##### Up stroke

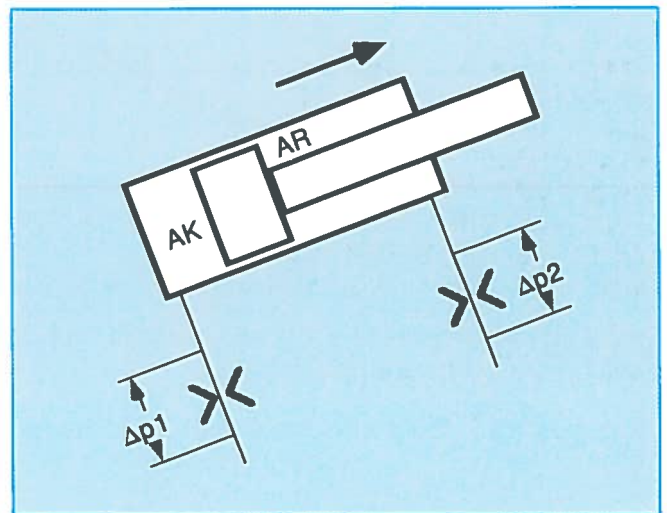


Fig. 16 Up stroke

Fig. 15 Flow/current curve for  $150 \text{ dm}^3/\text{min}$  nominal flow at  $10 \text{ daN/cm}^2$  valve pressure drop

The following values are obtained during the up stroke

Acceleration

$$\Delta p_1 = (19.64 \cdot 55 - 9100/10)/(19.64 + 9.45)$$

$$\approx 6 \text{ daN/cm}^2$$

$$p_V = 2 \cdot \Delta p_1 \approx 12 \text{ daN/cm}^2$$

Constant velocity

$$\Delta p_1 = (19.64 \cdot 55 - 3500/10)/(19.64 + 9.45)$$

$$\approx 25 \text{ daN/cm}^2$$

$$p_V = 2 \cdot \Delta p_1 \approx 50 \text{ daN/cm}^2$$

Deceleration

$$\Delta p_1 = (19.64 \cdot 55 + 2100/10)/(19.64 + 9.45)$$

$$\approx 45 \text{ daN/cm}^2$$

$$p_V = 2 \cdot \Delta p_1 \approx 90 \text{ daN/cm}^2$$

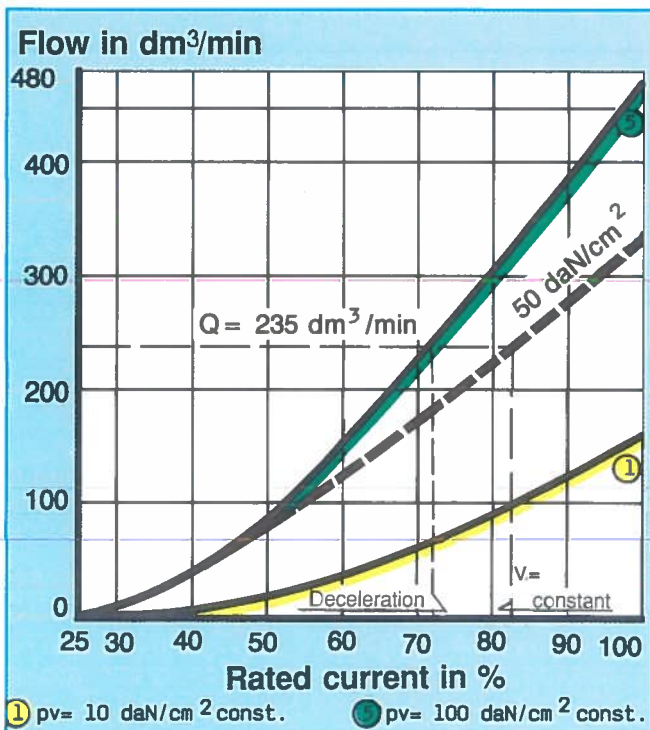


Fig. 17 Flow/current curve for 150 dm<sup>3</sup>/min nominal flow at 10 daN/cm<sup>2</sup> valve pressure drop

During the transition from constant velocity to deceleration of the system, a greater percentage change in the spool stroke is necessary at this pump pressure than at  $p_P = 100 \text{ daN/cm}^2$ .

This proportional change in the spool stroke at the proportional valve requires a correspondingly longer positioning time. The increase in deceleration is only slight during this phase.

### Calculation of the pressure drop at the throttle edges of 4-way proportional valves, taking into consideration the cylinder area ratio and the control land opening ratio at the valve.

Proportional directional valves are available as standard with a control land ratio  $F = 1:1$  and  $F = 2:1$ .

When calculating the pressure drop at the control lands,  $F$  must be considered corresponding to the cylinder area ratios.

The relationship

$$A_A > A_B \rightarrow X = F$$

$$A_B > A_A \rightarrow X = 1/F$$

is used in the following calculations depending on whether the area  $A_A$  is greater than  $A_B$  or vice versa.

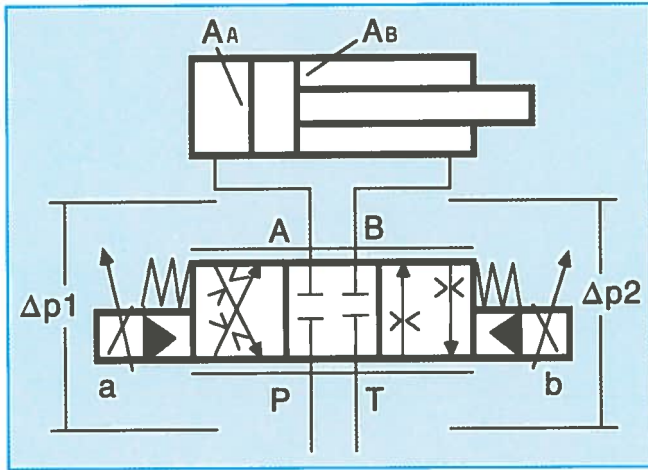


Fig. 18

The following applies for "extend cylinder" (drive side A)

Flow  $P \rightarrow A$

$$Q_A = \alpha \cdot A_{SA} \cdot \sqrt{\Delta p_1}$$

$A_{SA}$  = Clear throttle opening at the control land of the proportional valve from  $P \rightarrow A$

$$\rightarrow A_{SA} = Q_A / (\alpha \cdot \sqrt{\Delta p_1})$$

Control land ratio at proportional valve

$$A_{SA} / A_{SB} = X$$

$A_{SB}$  = Clear throttle opening at the control land of the proportional valve  $B \rightarrow T$

$$\rightarrow A_{SB} = A_{SA} / X$$

For  $A_{SA}$  is used  $Q_A / (\alpha \cdot \sqrt{\Delta p_1})$

$$\rightarrow A_{SB} = Q_A / (\alpha \cdot \sqrt{\Delta p_1} \cdot X)$$

Flow  $B \rightarrow T$

$$Q_B = \alpha \cdot A_{SB} \cdot \sqrt{\Delta p_2}$$

$$\rightarrow \sqrt{\Delta p_2} = Q_B / (\alpha \cdot A_{SB})$$

for  $A_{SB}$  is used  $Q_A / (\alpha \cdot \sqrt{\Delta p_1} \cdot X)$

$$\rightarrow \sqrt{\Delta p_2} = Q_B \cdot \alpha \cdot \sqrt{\Delta p_1} \cdot X / (\alpha \cdot Q_A)$$

The relationship of the flow with respect to the cylinder areas is as follows

$$Q_B / Q_A = A_B / A_A$$

used in  $\Delta p_2$

$$\rightarrow \Delta p_2 = A_B^2 / A_A^2 \cdot \Delta p_1 \cdot X^2$$

Equilibrium of forces during cylinder extension

$$F_G / 10 = A_A \cdot [(p_P - \Delta p_V) - \Delta p_1] - A_B \cdot \Delta p_2$$

for  $\Delta p_2$  is used  $A_B^2 / A_A^2 \cdot \Delta p_1 \cdot X^2$

$$F_G / 10 = A_A \cdot (p_P - \Delta p_V) - A_A \cdot \Delta p_1 - A_B \cdot \Delta p_1 \cdot X^2 \cdot A_B^2 / A_A^2$$

The equation multiplied by  $A_A^2$

$$A_A^2 \cdot F_G / 10 = A_A^3 \cdot (p_P - \Delta p_V) - A_A^3 \cdot \Delta p_1 - \Delta p_1 \cdot X^2 \cdot A_B^3$$

$$\Delta p_1 = \frac{A_A^2 \cdot [A_A \cdot (p_P - \Delta p_V) - F_G / 10]}{(A_A^3 + A_B^3 \cdot X^2)}$$

For retract cylinder (drive side B)

$$-F_G / 10 = A_B \cdot [(p_P - \Delta p_V) - \Delta p_2] - A_A \cdot \Delta p_1$$

$$-F_G / 10 = A_B \cdot [(p_P - \Delta p_V) - A_B^2 / A_A^2 \cdot \Delta p_1 \cdot X^2] - A_A \cdot \Delta p_1$$

The equation multiplied by  $A_A^2$

$$-A_A^2 \cdot F_G / 10 = A_A^2 \cdot A_B \cdot (p_P - \Delta p_V) - A_B^3 \cdot X^2 \cdot \Delta p_1 - A_A^3 \cdot \Delta p_1$$

$$\Delta p_1 = \frac{A_A^2 \cdot [A_B \cdot (p_P - \Delta p_V) + F_G / 10]}{(A_A^3 + A_B^3 \cdot X^2)}$$

$$\Delta p_2 = \Delta p_1 = A_B^2 \cdot X^2 / A_A^2$$

The total pressure drop at valve and the corresponding flow can be determined for cylinder extend and retract from the pressure drop at the individual control lands.

The smallest  $\Delta p$  must always be selected for the greatest flow.

Example:

Total valve  $\Delta p = p_V = 2 \cdot \Delta p_1$  for  $Q = \dots$

[dm<sup>3</sup>/min].

## Accuracy of Deceleration Distance for Time-Dependent Deceleration

The trend is moving towards higher and higher travel speeds in order to reduce idle times. However, this is only effective when the deceleration distance remains constant during all operating conditions.

If the deceleration distance varies, a longer creep distance must be included in the design.

**This costs time.**

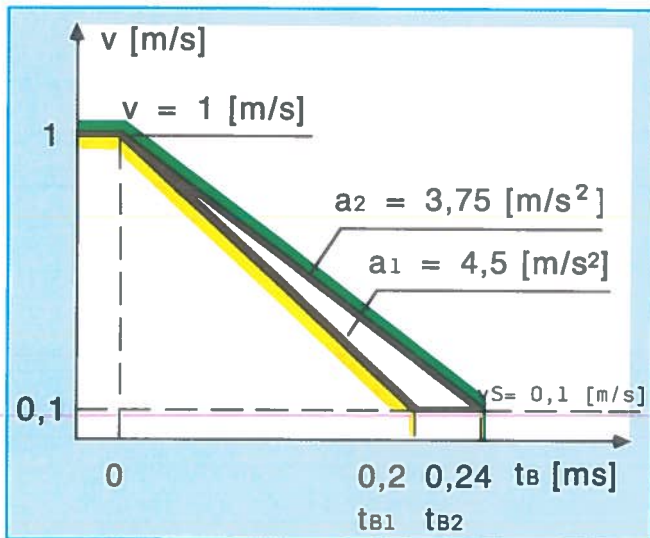


Fig. 19 Deceleration distance with respect to changed deceleration time

### Which factors can change the deceleration distance?

#### a) Change in the deceleration time (ramp)

In this case, proportional hydraulics with its electronic ramp offers a distinct advantage compared to a hydraulic switching time. A well formed electronic ramp is not subject to any marked changes as the result of the influences of temperature.

In order to fully utilize this advantage, care must be taken to ensure that the ramp time is not selected too short, i.e. sufficient distance is maintained for the natural hydraulic operating time of the proportional device.

#### Rule of thumb

Min. ramp time  $> 2 \times$  natural hydraulic operating time

The natural hydraulic operating time is specified in the data sheets for the proportional devices (stepped response).

The effect of the change in ramp time can be seen from the diagram Fig. 19.

The calculation shows that, at a deceleration of  $v = 1$  m/s, the creep stroke  $s_S$  has changed by 22 mm with respect to a creep speed of  $v_S = 0.1$  m/s. The time

necessary for this rate of deceleration is  $t_S = 220$  ms.

$$s_{B1} = (v + v_S)/2 \cdot t_{B1} = (1 + 0.1)/2 \cdot 0.2 = 0.11 \text{ [m]} = 110 \text{ [mm]}$$

$$s_{B2} = (v + v_S)/2 \cdot t_{B2} = (1 + 0.1)/2 \cdot 0.24 = 0.132 \text{ [m]} = 132 \text{ [mm]}$$

$$\Delta s_B = 22 \text{ [mm]}$$

$t_S$  for  $s_S = 22$  [mm] at  $v_S = 0.1$  [m/s]

$$t_S = s_S / v_S = (22 \cdot 10^{-3}) / 0.1 = 0.22 \text{ [s]} = 220 \text{ [ms]}$$

#### b) Varying lost times in the electrical signal

Processing of the electrical signals from the limit switch to the signal input on the electronic card should be short and must not be subject to changes in time.

At 1 (m/s), a change in the idle time of 10 (ms) results in the stroke changing by 10 (mm). A time  $t_S = 100$  (ms) is therefore required for a creep speed of 0.1 (m/s).

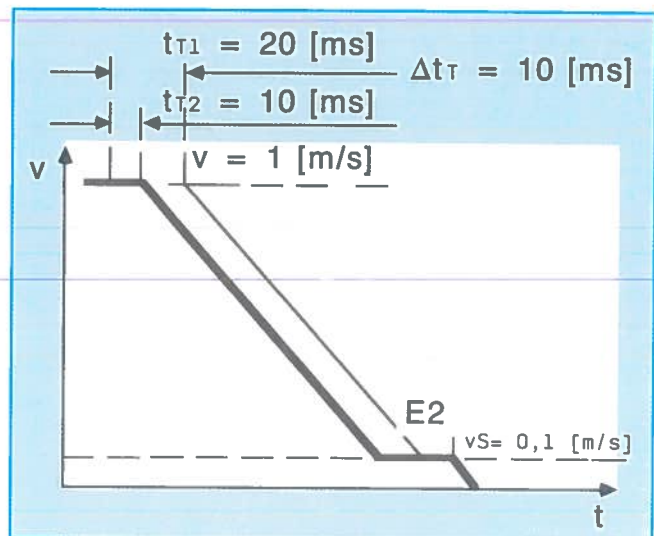


Fig. 20 Deceleration path with respect to a change in the idle time.

Distance in 10 [ms] at  $v = 1$  [m/s] = 10 [mm]

Time for 10 [mm] at  $v_S = 0.1$  [m/s] = 100 [ms]

#### c) Change in the velocity $v$ as the result of varying $\Delta p$ at the control lands of the proportional valve

Viscosity effects the flow and therefore the velocity at the driven unit.

The control lands of the proportional devices are designed as sharp edged orifices in order to achieve a low viscosity effect.

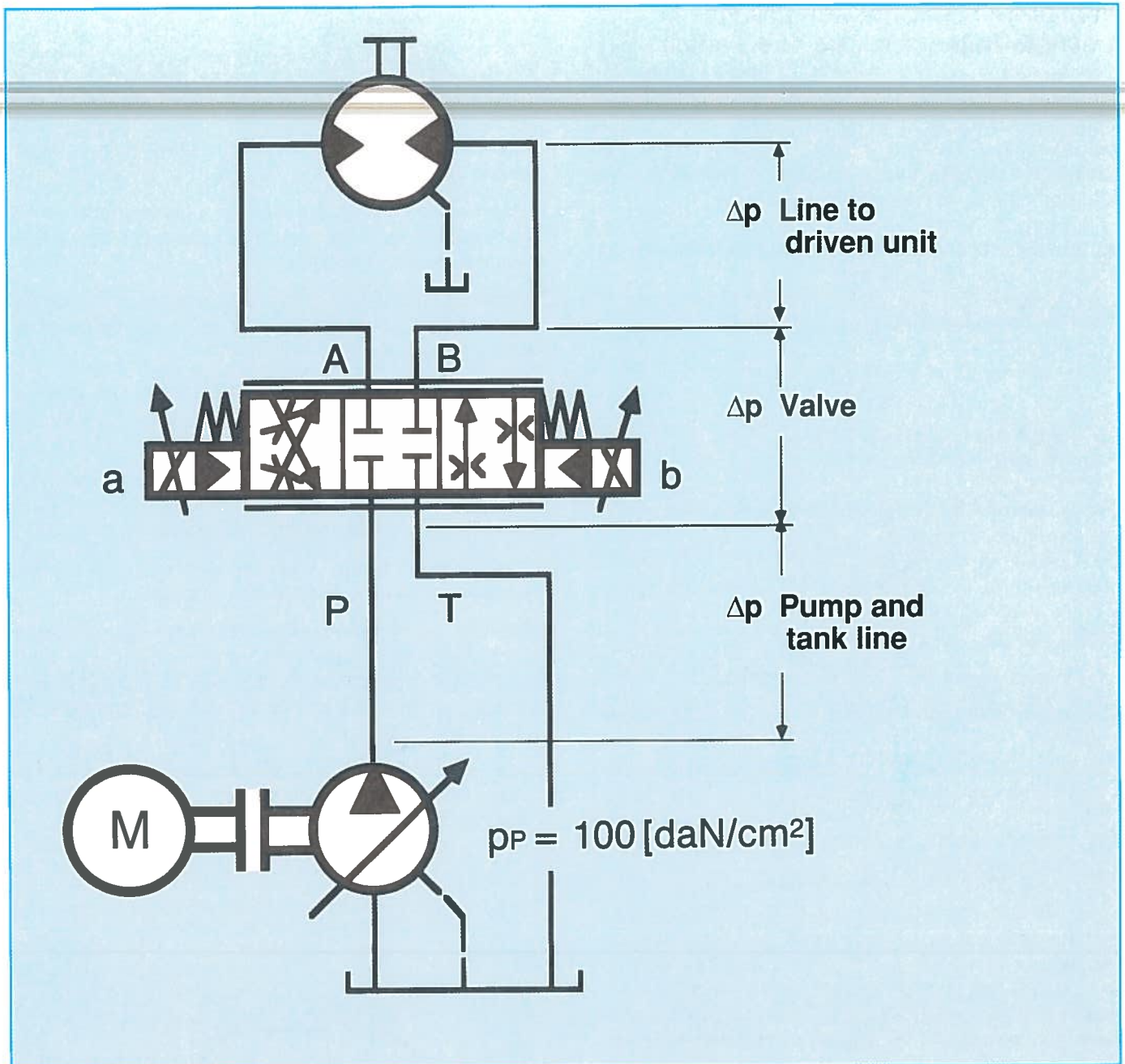


Fig. 21 Pressure drop in a hydraulic system

Measurements in hydraulic systems show that in the case of a change in temperature, the changes in  $\Delta p$  in the pipes, screw fittings, control manifolds are considerably greater as a percentage than the changes in  $\Delta p$  at the flow control unit itself.

These changes in  $\Delta p$  naturally result in a change in flow.

The effect on the deceleration distance is explained by means of an example.

$\Delta p$  Line to driven unit = 4 [daN/cm<sup>2</sup>] at 50 °C

$\Delta p$  Line to driven unit = 6 [daN/cm<sup>2</sup>] at 20 °C

$\Delta p$  Pump line = 5 [daN/cm<sup>2</sup>] at 50 °C

$\Delta p$  Pump line = 8 [daN/cm<sup>2</sup>] at 20 °C

Change in  $\Delta p$  at proportional valve = 5 [daN/cm<sup>2</sup>]

## Variations in velocity with changes in viscosity in the pipelines of a system

The total pressure drop of 55 daN/cm<sup>2</sup> measured at the valve is obtained at an oil temperature of 50 °C as a result of the installed pump pressure  $p_p = 100$  (daN/cm<sup>2</sup>) and the torque obtained at the motor. The maximum measured velocity is  $v = 1.3$  (m/s).

At low oil temperature, the total pressure drop at the valve is only  $\Delta p = 50$  (daN/cm<sup>2</sup>).

The travel speed at 20 °C is obtained from

$$20\text{ }^{\circ}\text{C} \rightarrow v = v \cdot \sqrt{\Delta p_1 / \Delta p_2} = 1.3 \cdot \sqrt{50/55} = 1.24 \quad [\text{m/s}]$$

$\Delta p_1$  = Pressure drop at valve at 20 °C

$\Delta p_2$  = Pressure drop at valve at 50 °C

The acceleration and deceleration distances are at

$$50\text{ }^{\circ}\text{C} \rightarrow s_B = \frac{v^2}{(2 \cdot a)} \cdot 10^3 = \frac{1.3^2}{(2 \cdot 2)} \cdot 10^3 = 422.5 [\text{mm}]$$

$$20\text{ }^{\circ}\text{C} \rightarrow s_B = \frac{v^2}{(2 \cdot a)} \cdot 10^3 = \frac{1.24^2}{(2 \cdot 2)} \cdot 10^3 = 422.5 [\text{mm}]$$

Change in deceleration distance  $\approx 38.5$  [mm]

At  $\Delta p = 50$  (daN/cm<sup>2</sup>), the pressure drop at the proportional valve is relatively high.

For energy reasons, it seems appropriate to reduce the system pressure.

However, with regard to the change in the deceleration distance this is not advisable as shown in the following calculation.

Under the assumption that the minimum total pressure drop at valve up to 20 °C  $\Delta p = 10$  (daN/cm<sup>2</sup>) and corresponding to the pipe losses at 50 °C  $\Delta p = 15$  (daN/cm<sup>2</sup>) the following is derived

$$50\text{ }^{\circ}\text{C} \rightarrow v = 1.3 \quad [\text{m/s}]$$

$$20\text{ }^{\circ}\text{C} \rightarrow v = v \cdot \sqrt{\Delta p_1 / \Delta p_2} = 1.3 \cdot \sqrt{10/15} = 1.06 \quad [\text{m/s}]$$

$\Delta p_1$  = Pressure drop at valve at 20 °C

$\Delta p_2$  = Pressure drop at valve at 50 °C

The acceleration and deceleration distances are at

$$50\text{ }^{\circ}\text{C} \rightarrow s_B = \frac{v^2}{(2 \cdot a)} \cdot 10^3 = \frac{1.3^2}{(2 \cdot 2)} \cdot 10^3 = 422.5 [\text{mm}]$$

$$20\text{ }^{\circ}\text{C} \rightarrow s_B = \frac{v^2}{(2 \cdot a)} \cdot 10^3 = \frac{1.06^2}{(2 \cdot 2)} \cdot 10^3 = 281.6 [\text{mm}]$$

Change in deceleration distance  $\approx 150$  [mm]

This change in the deceleration does not occur in a load compensated control with pressure compensator since the pressure drop at the throttling point is maintained constant.

The pressure drop at the valve in a throttle control must be selected higher to ensure the spread of the deceleration distance is kept small.

It should be noted that this additional  $\Delta p$  is low at low acceleration, while the amount must be assumed as being higher for high acceleration.

This energy loss as the result of throttling and which at first glance appears to be high only occurs briefly due to the high velocity of the driven unit.

The pressure compensator cannot be used for dynamic reasons in the case of high travel speeds (reference value  $> 1$  m/s) and fast acceleration sequences.

The throttle control, however, results in excessively large changes in the deceleration distance.

The use of an electronic control with distance-dependent deceleration provides a considerably better result at this high rate of deceleration with regard to the consistency of the deceleration distance and therefore the consistency of the travel time (Fig. 22).

The proportional valve is closed dependent on the distance travelled by the driven unit. The travel distance may be measured by analogue or digital methods.

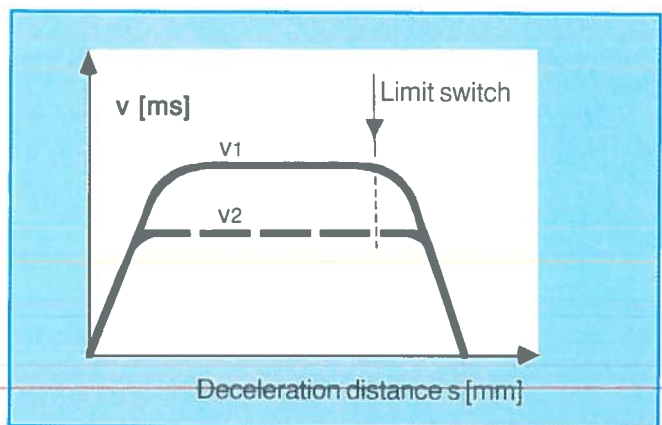


Fig. 22

### Calculation of Cylinder and Motor Dimensions for Control System with 4-Way Proportional Valves

A single set pump pressure is assumed for the calculation

#### a) Linear movement through cylinder drive

The following data are required

Mass moved per cylinder	m	[kg]
Static load per cylinder	F <sub>St</sub>	[N]
Static force per cylinder	F <sub>K</sub>	[N]
Frictional force per cylinder	F <sub>R</sub>	[N]
Cylinder velocity	v	[m/s]
Required acceleration time	t <sub>B</sub>	[s]
Pump pressure	p <sub>P</sub>	[daN/cm <sup>2</sup> ]
Δp losses in pipe	Δp <sub>V</sub>	[daN/cm <sup>2</sup> ]

#### b) Rotary movement through hydraulic motor drive

The following data are required

Moment of inertia at output	J	[kgm <sup>2</sup> ]
Static load torque at output	M <sub>St</sub>	[Nm]
Static torque to produce force at output	M <sub>K</sub>	[Nm]
Frictional torque at output	M <sub>C</sub>	[Nm]
Moment of inertia of motor and gear mechanism	J <sub>M</sub>	[kgm <sup>2</sup> ]
Transmission ratio	i	
Motor speed/output speed	= n <sub>M</sub> /n	
Speed at motor	n <sub>M</sub>	[rpm]
Speed at output	n	[rpm]
Required acceleration time	t <sub>B</sub>	[s]
Pump pressure	p <sub>P</sub>	[daN/cm <sup>2</sup> ]
Δp losses in pipe	Δp <sub>V</sub>	[daN/cm <sup>2</sup> ]

### Determining the Effective Cylinder Area (or Capacity of Hydraulic Motor) Required for Throttle Control (without Load Compensation)

Experience has shown that a cylinder or hydraulic motor is well dimensioned when the available pressure

$$[p_P - \Delta p_V]$$

is distributed in

- 1/3 for load
- 1/3 for acceleration
- 1/3 for velocity.

This means, that taking the load as 1/3 of the total pressure, only 1/2 of  $[(p_P - \Delta p_V) - p_{St}]$  is used for the deceleration of the mass, or of the inertia of the rotating parts. If this is not so, the spool movement of the proportional valve must be too great when changing from a constant speed to the deceleration phase. The pressure required by the load seldom amounts to 1/3 of the total. For this reason, it is always good practice to subtract the actual load pressure from the av. pressure and to determine the dimensions of the cylinder or motor in accordance with the above specified formula.

In the case of the cylinder drive, the effective area for acceleration or deceleration is calculated as follows:

$$\begin{aligned} 1/10 \cdot a \cdot m &= \Delta p \cdot A_W & A_W &= \text{effective area [cm}^2\text{]} \\ \Delta p &= 1/2 \cdot [(p_P - \Delta p_V) - p_{St}] & a &= \text{Acceleration [m/s}^2\text{]} \\ p_{St} &= (F_{St} + F_R) / (A_W \cdot 10) & \Delta p &= \text{Eff. pressure [daN/cm}^2\text{]} \\ a &= v/t_B \end{aligned}$$

$$\begin{aligned} &\rightarrow v/t_B \cdot m/10 = \\ &= 1/2 \cdot [(p_P - \Delta p_V) - (F_{St} + F_R) / (A_W \cdot 10)] \cdot A_W \end{aligned}$$

$$A_W \geq \frac{2 \cdot [10 \cdot (p_P - \Delta p_V)] \cdot [m \cdot v/t_B + 1/2 \cdot (F_{St} + F_R)]}{[cm^2]}$$

The working or pressing force F<sub>K</sub> is not effective during the acceleration or deceleration phase.

#### Note:

If a percentage force is to be effective during the acceleration phase, then it must be added to the static force F<sub>K</sub>.

The effective area for constant travel speed and max. force F<sub>K</sub> is

$$A_W = (F_{St} + F_K + F_R) / [10 \cdot (p_P - \Delta p_V - 10)] \quad [cm^2]$$

"10" → Minimum pressure drop at proportional valve

## Design Criteria for Open Loop Control with Proportional Valves

The largest effective area obtained from the two calculations defines the cylinder dimensions.

If the effective area and the acceleration time are known, the required pump pressure can be correspondingly calculated.

For acceleration

$$p_P = \frac{2 \cdot m \cdot v}{(t_B \cdot 10 \cdot A_W)} + \Delta p_v + \frac{(F_{St} + F_R)}{(10 \cdot A_W)} \quad [\text{daN/cm}^2]$$

At constant velocity and max. force

$$p_P = \frac{(F_{St} + F_K + F_R)}{(10 \cdot A_W)} + \Delta p_v + 10 \quad [\text{daN/cm}^2]$$

"10" → Minimum pressure drop at proportional valve

If the effective area and the pump pressure are known, the acceleration time can be calculated as follows

$$t_B = \frac{(2 \cdot m \cdot v)}{[10 \cdot A_W \cdot (p_P - \Delta p_v) - (F_{St} + F_R)]} \quad [\text{s}]$$

The capacity for a hydraulic motor drive in throttle control for acceleration and deceleration is calculated as follows

$$J_G / i^2 \cdot \varepsilon = (V_G \cdot \Delta p) / (20 \cdot \pi)$$

$$\Delta p = 1/2 \cdot [(p_P - \Delta p_v) - p_S]$$

$$\varepsilon = \omega / t_B$$

$$\omega = \pi \cdot n \cdot i / 30$$

$$p_S = [(M_S + M_C) \cdot 20 \cdot \pi] / (i \cdot V_G)$$

$$J_G / i^2 \cdot \pi \cdot n \cdot i / (30 \cdot t_B) = V_G / (20 \cdot \pi) \cdot 1/2 \{ (p_P - \Delta p_v) - [(M_S + M_C) \cdot 20 \cdot \pi] / (i \cdot V_G) \}$$

$$V_G = \frac{4 \cdot \pi / [(p_P - \Delta p_v) \cdot i] \cdot [J_G \cdot n \cdot \pi / (3 \cdot t_B) + 5 \cdot (M_S + M_C)]}{\quad} \quad [\text{cm}^3]$$

$$\Delta p = \text{Required pressure difference for acceleration} \quad [\text{daN/cm}^2]$$

$$J_G = \text{Total moment of inertia at output} \quad [\text{kgm}^2]$$

$$i = \text{Motor-n/output-n}$$

$$V_G = \text{Motor capacity} \quad [\text{cm}^3]$$

$$\omega = \text{Angular velocity} \quad [\text{rad/s}]$$

$$\varepsilon = \text{Angular acceleration} \quad [\text{rad/s}^2]$$

### Note:

If a useful torque  $M_K$  is effective during the acceleration phase then this must be added to the load torque  $M_S$ .

For constant speed and maximum useful torque  $M_K$

$$V_G = \frac{(M_S + M_K + M_C) \cdot 20 \cdot \pi}{[i \cdot (p_P - \Delta p_v - 10)]} \quad [\text{cm}^3]$$

"10" → Minimum pressure drop at proportional valve

The maximum capacity determines the selection of the hydraulic motor.

If the capacity and the acceleration time are known, the required pump pressure can be correspondingly calculated.

For acceleration

$$p_P = \frac{J_G \cdot \pi^2 \cdot n \cdot 4 / (3 \cdot i \cdot V_G \cdot t_B) + \Delta p_v + [(M_S + M_C) \cdot 20 \cdot \pi] / (i \cdot V_G)}{\quad} \quad [\text{daN/cm}^2]$$

For constant speed and max. torque  $M_K$

$$p_P = \frac{[(M_S + M_K + M_C) \cdot 20 \cdot \pi] / (i \cdot V_G) + \Delta p_v + 10}{\quad} \quad [\text{daN/cm}^2]$$

"10" → Minimum pressure drop at proportional valve

If the volume and the pump pressure are known, the acceleration time can be calculated

$$t_B = \frac{4/3 \cdot J_G \cdot n \cdot \pi^2 [i \cdot V_G \cdot (p_P - \Delta p_v) - 20 \cdot \pi \cdot (M_S + M_C)]}{\quad} \quad [\text{s}]$$

### Determining the effective cylinder area required for load compensated controls

In the case of a load compensated control, the complete pressure ( $p_p - \Delta p_v$ ) is available less  $\Delta p$  of the pressure compensator (8 daN/cm<sup>2</sup>) and less the pressure drop at the throttle edge - consumer to tank (8 daN/cm<sup>2</sup>).

The effective area can be calculated for acceleration or deceleration

$$A_W \geq \frac{1/10 \cdot [(F_{St} + F_R) + v \cdot m/t_B]}{(p_p - \Delta p_v - 16)} \quad [\text{cm}^2]$$

For constant velocity and maximum force  $F_K$

$$A_W \geq \frac{(F_{St} + F_K + F_R)}{[10 \cdot (p_p - \Delta p_v - 16)]} \quad [\text{cm}^2]$$

"16" → Minimum pressure drop at proportional valve + at pressure compensator

The maximum effective area  $A_W$  determines the cylinder dimensions.

If the effective area and the acceleration time are known, the required pump pressure can be correspondingly calculated,

for acceleration

$$p_p = \frac{m \cdot v / (t_B \cdot 10 \cdot A_W) + \Delta p_v + 16 + (F_{St} + F_R) / (10 \cdot A_W)}{[\text{daN/cm}^2]}$$

for constant velocity and max. force  $F_K$

$$p_p = \frac{(F_{St} + F_K + F_R) / (10 \cdot A_W) + \Delta p_v + 16}{[\text{daN/cm}^2]}$$

"16" → Minimum pressure drop at proportional valve + at pressure compensator

If the effective area and the pump pressure are known, the acceleration time can be calculated

$$t_B = \frac{m \cdot v / [10 \cdot A_W \cdot (p_p - \Delta p_v - 16) - (F_{St} + F_R)]}{[\text{s}]}$$

In the case of a hydraulic motor drive, the capacity is calculated for load compensated control of acceleration and deceleration

$$V_G = \frac{2 \cdot \pi \cdot [(p_p - 16 - \Delta p_v) \cdot i] \cdot [J_G \cdot n \cdot \pi (3 \cdot t_B) + 10 \cdot (M_S + M_K)]}{[\text{cm}^3]}$$

For constant velocity and maximum "work" torque  $M_K$

$$V_G = \frac{(M_S + M_K + M_C) \cdot 20 \cdot \pi}{[i \cdot (p_p - \Delta p_v - 16)]} \quad [\text{cm}^3]$$

"16" → Minimum pressure drop at proportional valve + at pressure compensator

If the capacity and the acceleration time are known, the pump pressure required for acceleration can be calculated

$$p_p = \frac{J_G \cdot \pi^2 \cdot n \cdot 2 \cdot (3 \cdot i \cdot V_G \cdot t_B) + \Delta p_v + 16}{+ [(M_S + M_C) \cdot 20 \cdot \pi] / (i \cdot V_G)} \quad [\text{daN/cm}^2]$$

$$p_p = \frac{[(M_S + M_K + M_C) \cdot 20 \cdot \pi]}{[(i \cdot V_G) + \Delta p_v + 16]} \quad [\text{daN/cm}^2]$$

"16" → Minimum pressure drop at proportional valve

If the effective area and the pump pressure are known, the acceleration time can be correspondingly calculated

$$t_B = \frac{(2/3 \cdot J_G \cdot n \cdot \pi^2) / [i \cdot V_G \cdot (p_p - \Delta p_v - 16) - 20 \cdot \pi \cdot (M_S + M_C)]}{[\text{s}]}$$

### Calculation and Effect of the Natural Frequency of Hydraulic Systems

As already stated, the natural frequency is a measure for the stability of the drive and the minimum possible acceleration time.

Various parameters such as mechanical friction and oil viscosity must be known to calculate the precise natural frequency of a system.

These parameters are often not known in the planning phase. In practical applications, however, it is sufficient to calculate the undamped natural frequency and to derive "reference values" from the results.

The comparison with the natural frequency of a mechanical spring/mass system is used to facilitate understanding with regard to calculation of the undamped natural frequency of a hydraulic system.

### Natural Frequency for Double Rod Cylinder without Damping

- $A_W$  = Effective piston area  
 $h$  = Cylinder stroke  
 $V_{L1}$  =  $V_1$  = Pipe volume  
 $V_{L2}$  =  $V_2$  = Pipe volume  
 $V_A$  =  $V_B$  = Cylinder volume  
 $V_3, V_4$  = Oil volume between control valve and cylinder ( $V_1 + V_A$ )  
 $E_{Oil}$  = Modulus of elasticity of oil

### Overall Spring Constant

$$\begin{aligned}
 C_{overall} &= C_1 + C_2 \\
 &= 2 \cdot [A_W^2 \cdot E_{Oil} / (V_1 + h/2 \cdot 10) \cdot A_W] \\
 &= 2 \cdot [A_W^2 \cdot E_{Oil} / (V_1 + V_A)] \\
 &= 2 \cdot (A_W^2 \cdot E_{Oil} / V_3)
 \end{aligned}$$

### Substituting the above in a spring/mass system

- $F$  = Spring force  
 $T$  = Oscillation period for 1 complete oscillation cycle  
 $s$  = Spring travel

### Oscillation of the above spring/mass system without damping

### Circuit frequency of spring/mass system

$$\omega_0 = \sqrt{C_{ges}/m} \text{ [1/s]}$$

### Natural frequency

$$f = 1/(2 \cdot \pi) \cdot \sqrt{C_{ges}/m} \text{ [Hz]}$$

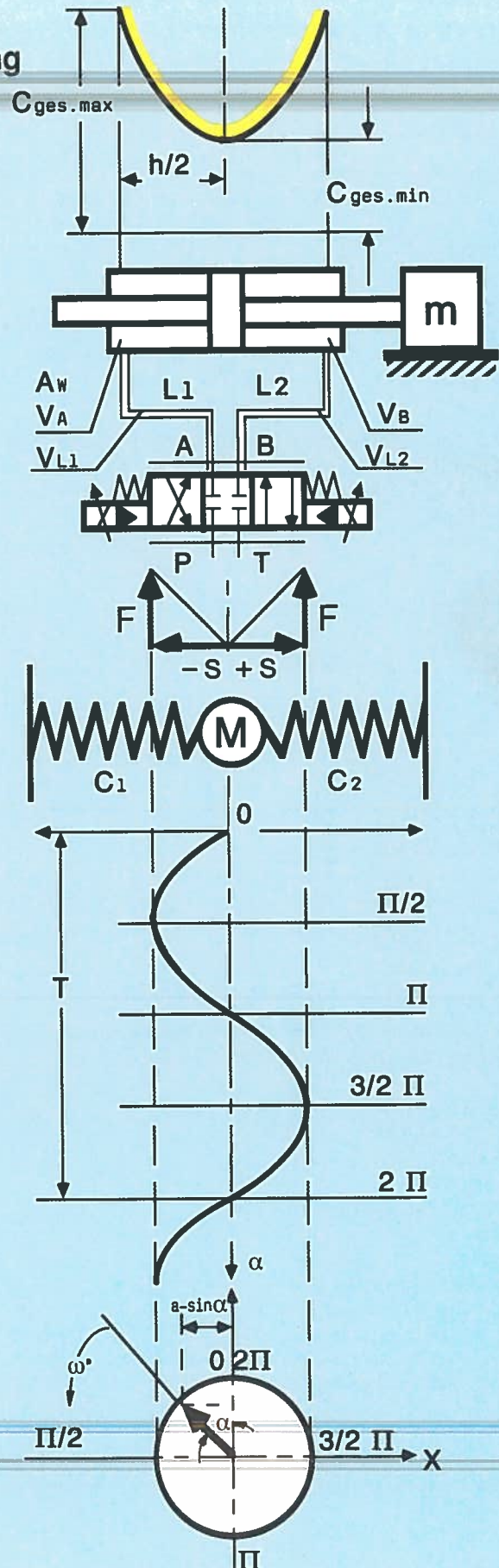


Fig. 23 Natural frequency for double rod cylinder without damping

# Determining Natural Frequency for a Hydraulic Cylinder

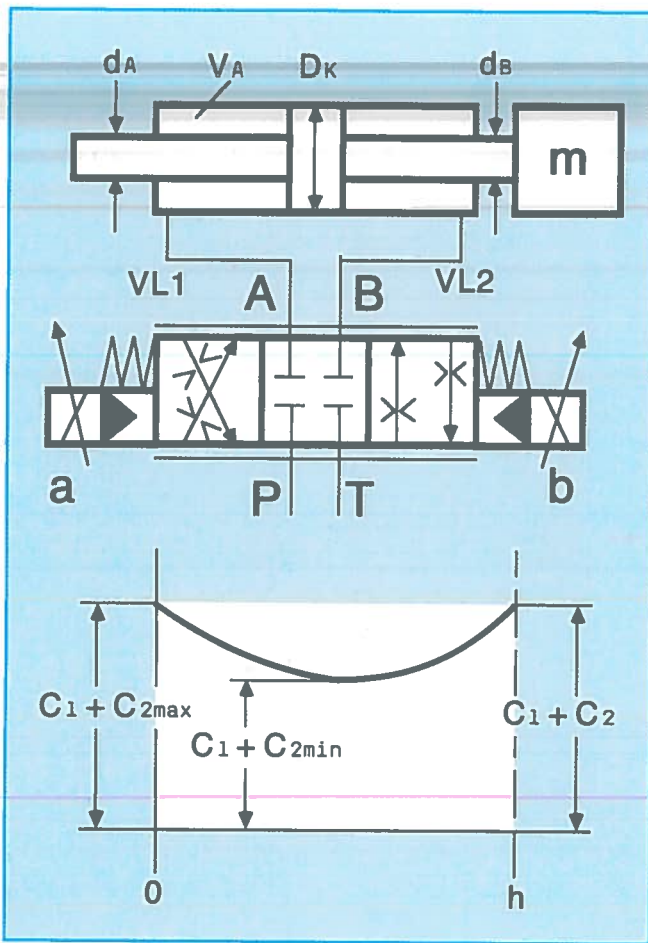


Fig. 24 Double rod cylinder

$$C_1 = A_A^2 \cdot E_{Oil} / [(A_A \cdot h/2/10) + V_{L1}] \quad [Nm]$$

$$C_2 = A_B^2 \cdot E_{Oil} / [(A_B \cdot h/2/10) + V_{L2}] \quad [Nm]$$

$$\omega_0 = \sqrt{(C_1 + C_2)/m} \quad [1/s]$$

The natural frequency is minimum in the mid position of the cylinder when the annulus area  $A_A = A_B$  and  $V_{L1} = V_{L2}$ .

Mass	m	[kg]
Cylinder stroke	h	[mm]
Cylinder stroke at min. natural frequency	$h_K$	[mm]
Piston area	$A_K$	[cm <sup>2</sup> ]
Annulus area	$A_R$	[cm <sup>2</sup> ]
Pipe volume on piston side	$V_{L1}$	[cm <sup>3</sup> ]
Pipe volume on annulus side	$V_{L2}$	[cm <sup>3</sup> ]
Modulus of elasticity = $1.4 \cdot 10^7$	$E_{Oil}$	[kg/cm <sup>2</sup> sec <sup>2</sup> ]
Spring constant on piston side	$C_1$	[N/m]
Spring constant on annulus side	$C_2$	[N/m]

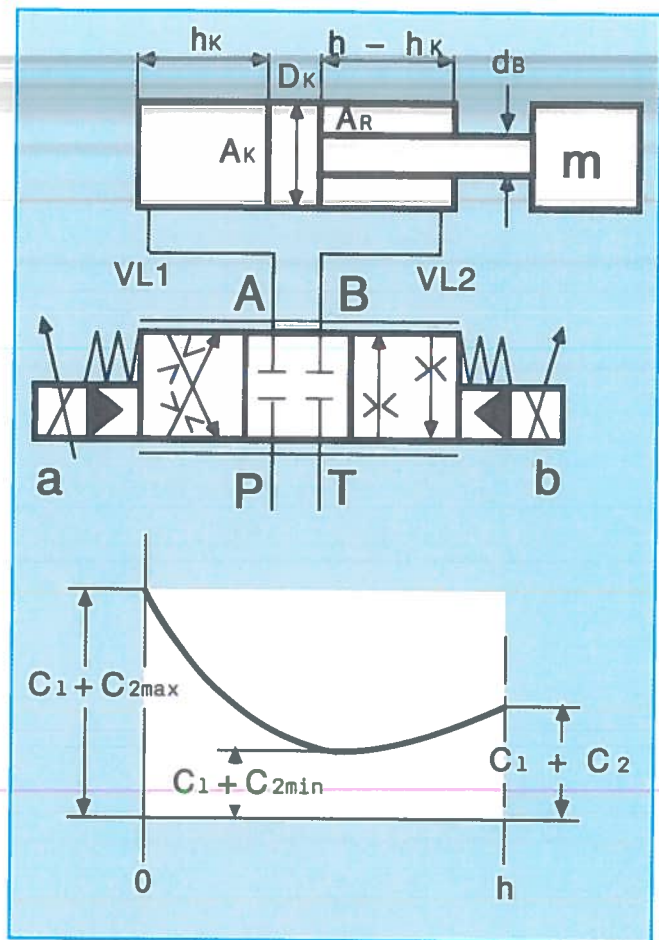


Fig. 25 Single rod cylinder

$$C_1 = A_K^2 \cdot E_{Oil} / (A_K \cdot h_K/10 + V_{L1}) \quad [Nm]$$

$$C_2 = A_R^2 \cdot E_{Oil} / [A_R \cdot (h - h_K)/10 + V_{L2}] \quad [Nm]$$

The piston position  $h_K$  at which the total spring load is at a minimum can be calculated as follows

$$(C_1 + C_2)_{\max} = A_K^2 \cdot E_{Oil} / V_{L1} + A_R^2 \cdot E_{Oil} / (V_{L2} + A_R \cdot h/10) \quad \text{for } h = 0$$

$$(C_1 + C_2)_{\max} = A_K^2 \cdot E_{Oil} / (V_{L1} + A_K \cdot h/10) + A_R^2 \cdot E_{Oil} / V_{L2} \quad \text{for } h = h$$

If the equation for  $(C_1 + C_2)$  is differentiated,  $(C_1 + C_2)_{\min}$  and the corresponding cylinder stroke  $h_K$  can be calculated.

$$h_K = [(A_R \cdot h/10 \sqrt{A_R^3 + V_{L1}} \sqrt{A_R^3 + V_{L2}} \sqrt{A_K^3}) / (1/\sqrt{A_R} + 1/\sqrt{A_K})] \cdot 10 \quad [mm]$$

$$\omega_0 = \sqrt{(C_1 + C_2)/m} \quad [1/s]$$

### Determining the natural frequency for a hydraulic cylinder with regenerative control circuit

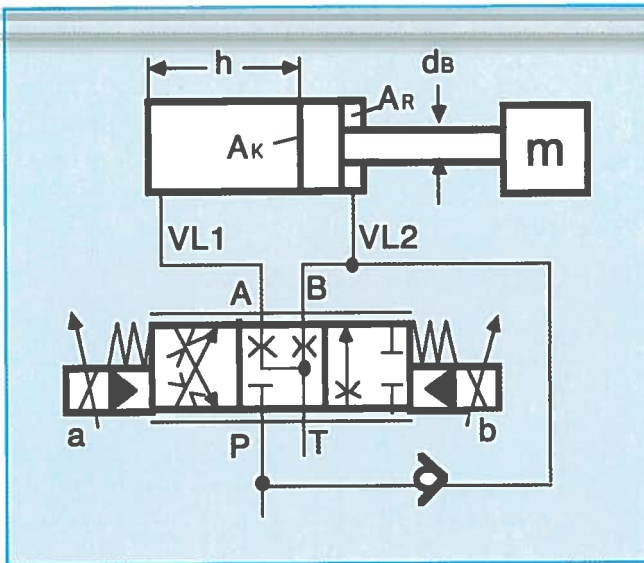


Fig. 26

$$C_1 = A_K^2 \cdot E_{Oil} / (A_K \cdot h/10 + V_{L1}) \quad [N/m]$$

$$\omega_0 = \sqrt{C_1/m} \quad [1/s]$$

In the case of a regenerative control circuit, there is no spring constant  $C_2$  on the annulus side  $A_R$  during cylinder extension.

#### Reason

The annulus side is under constant pressure  $p_p$ . Outer forces which act on the cylinder do not result in a pressure increase on this side of the cylinder - no increase in counterforce in this annulus chamber of the cylinder.

The lowest spring constant  $C_1$  and therefore the lowest natural frequency is obtained at piston stroke  $h$ .

### Determining the natural frequency for a hydraulic cylinder with load compensated control

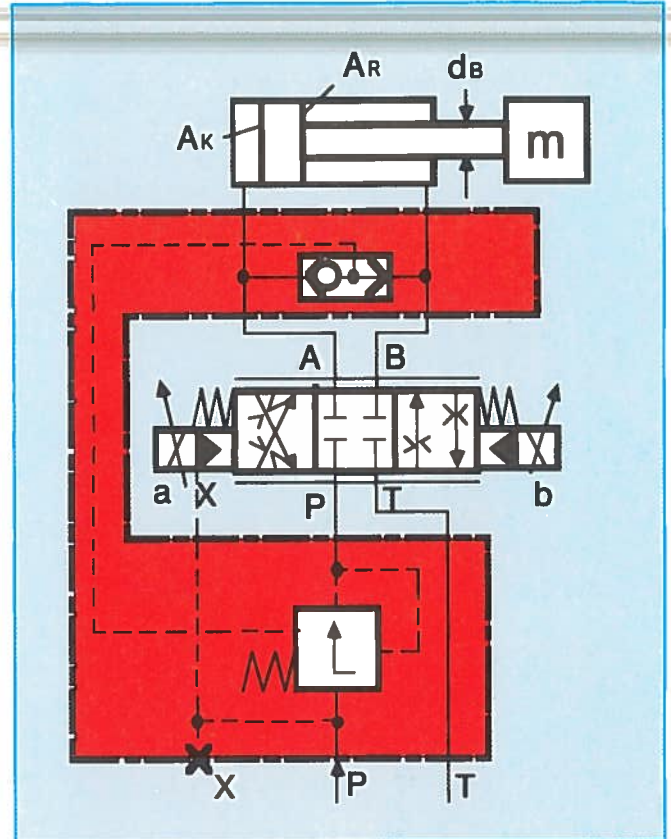


Fig. 27

$$C_2 = A_R^2 \cdot E_{Oil} / (A_R \cdot h/10 + V_{L2}) \quad [N/m]$$

$$\omega_0 = \sqrt{C_2/m} \quad [1/s]$$

Also in the case of load compensated controls, the spring constant can be expected only on one side of the cylinder.

The side which is not load compensated is under constant backpressure of the throttle edge for the outgoing oil (for meter-in pressure compensator).

Outer forces do not result in an increase in pressure and therefore an increase in force on this side.

The lowest spring constant and therefore the lowest natural frequency are obtained when the cylinder is retracted.

Determining natural frequency  
for hydraulic drives with hydraulic motors

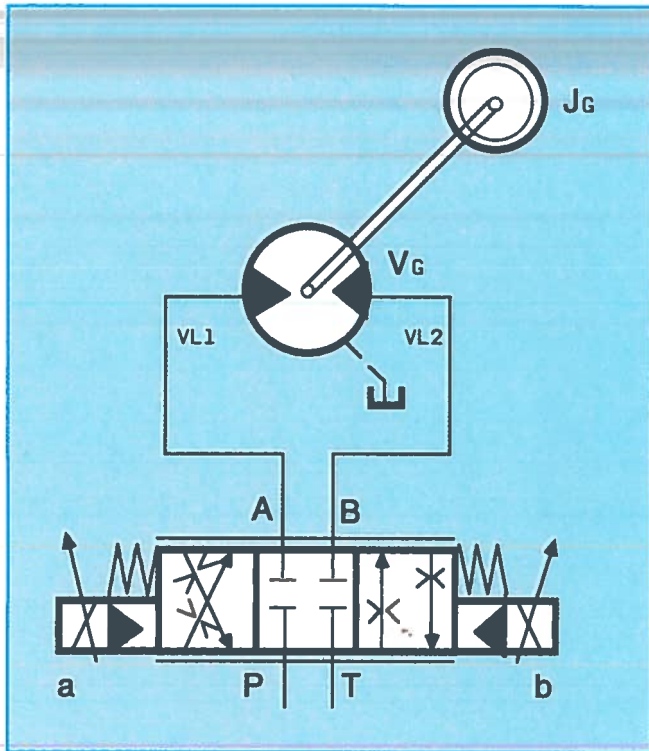


Fig. 28

$$C_1 = \frac{[V_G / (2 \cdot \pi)]^2 \cdot E_{Oil}}{[(V_G / 2 + V_{L1}) \cdot 10^4]} \quad [\text{Nm/rad}]$$

$$C_2 = \frac{[V_G / (2 \cdot \pi)]^2 \cdot E_{Oil}}{[(V_G / 2 + V_{L2}) \cdot 10^4]} \quad [\text{Nm/rad}]$$

$$\omega_0 = \sqrt{(C_1 + C_2) / J_G} \quad [1/\text{s}]$$

Moment of inertia	$J_G$	$[\text{kgm}^2]$
Capacity hydraulic motor	$V_G$	$[\text{cm}^3/\text{U}]$
Pipe volume	$V_{L1}$	$[\text{cm}^3]$
	$V_{L2}$	$[\text{cm}^3]$
Modulus of elasticity $1.4 \cdot 10^7$	$E_{Oil}$	$[\text{kg/cm} \cdot \text{sec}^2]$

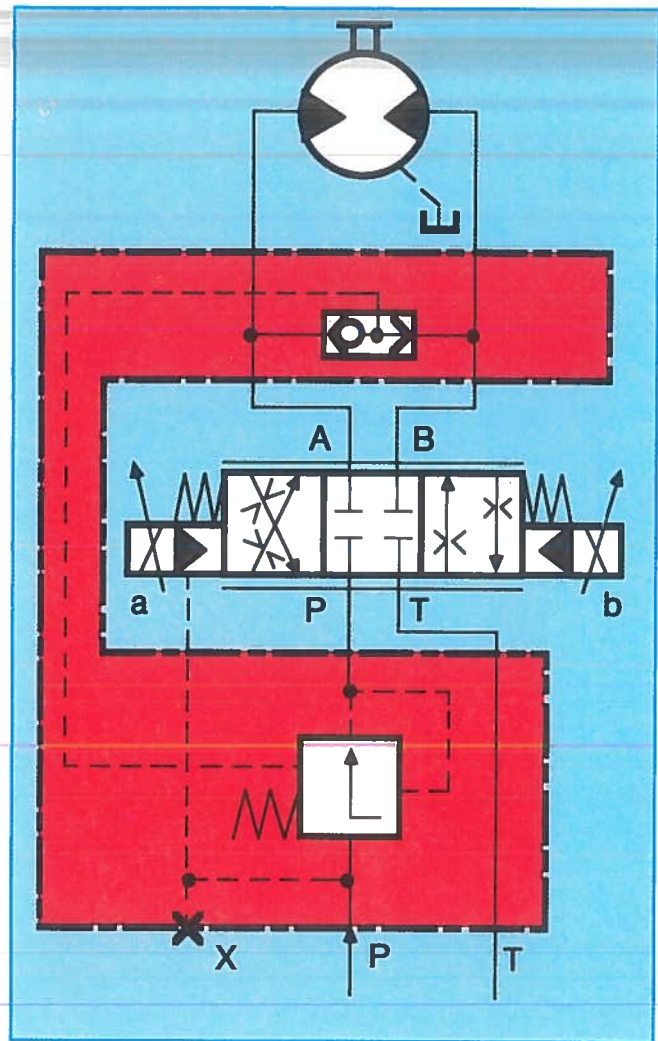


Fig. 29

In load compensated controls, the spring constant can be expected only on one side of the motor.

The side which is not load compensated is under constant backpressure of the throttle edge for the meter-in pressure compensator).

Outer forces do not result in an increase in pressure and therefore an increase in force on this side.

$$\omega_{0\min} = \sqrt{C_1 / J_G} \quad [1/\text{s}]$$

## Which Reference Values can be Derived from the Natural Frequency Calculated for Control Systems with Proportional Devices?

### a) Lowest System Natural Frequency

In control systems, the natural frequency should not drop below the following values

without load compensation	3 Hz = 18.84	[1/s]
with load compensaton	4 Hz = 25.13	[1/s]

In the case of smaller natural frequencies in the system, it is found that acceleration and deceleration procedures no longer function correctly due to the low system stiffness. In addition, stick-slip can be expected at low travel speeds.

In controls with load compensation these negative characteristics appear earlier since the pressure compensator also has a natural time characteristic. Throttle controls (without load compensation) have an additional damping effect and can smooth an uneven velocity progression at low frequencies of the system better.

An approximate velocity progression cannot be expected also in the throttle control when the difference between static and sliding friction (stick-slip) is great.

### b) Minimum Acceleration and Deceleration Time

A reference value for the acceleration and deceleration time can be derived from the natural frequency. For control with proportional directional and flow control valves this results in

$$t_B = 18/\omega_0 \quad [1/s]$$

$$\omega_0 = \text{undamped circuit frequency of system in} \quad [1/s]$$

For practical applications, this acceleration/ deceleration time is shown on *Page 28* in tabular form as a function of the circuit frequency.

The acceleration values  $a$  in  $(m/s^2)$  are also listed for various travel speeds.

Valve limit	Circuit frequency (undamped)	Circuit frequency (undamped)	Acceleration/ deceleration time (minimum)	For velocity v [m/s]			
	$\omega_0 [s^{-1}]$	f [Hz]	t <sub>b</sub> [s]	v = 0,5	v = 1	v = 1,5	v = 2
				the following acceleration/deceleration a in [ms] results			
Size 32	5	0,79	3,6	0,138	0,277	0,416	0,555
Size 25	10	1,59	1,85	0,277	0,55	0,833	1,11
Size 16	15	2,38	1,2	0,416	0,833	1,25	1,66
Size 10	20	3,18	0,9	0,555	1,11	1,66	2,22
	30	4,77	0,6	0,833	1,66	2,5	3,33
	40	6,37	0,45	1,111	2,22	3,33	4,44
	50	7,95	0,36	1,388	2,77	4,16	5,55
	60	9,54	0,3	1,666	3,33	5,0	6,66
	70	11,14	0,26	1,94	3,89	5,83	7,77
	80	12,73	0,225	2,22	4,44	6,66	8,88
	90	14,32	0,2	2,5	5,0	7,5	10,0
	100	15,91	0,18	2,77	5,56	8,33	11,11
	110	17,50	0,16	3,05	6,11	9,16	12,22
	120	19,09	0,15	3,33	6,66	10,0	
	130	20,69	0,138	3,61	7,22	10,83	
	140	22,28	0,128	3,88	7,78		
	150	23,87	0,12	4,16	8,33		
	160	25,46	0,1125	4,44	8,89		
	170	27,05	0,105	4,72	9,44		
	180	28,64	0,1	5,0	10,0		

### Note

The minimum acceleration/deceleration time can be defined by 3 characteristic variables:

1. Minimum acceleration/deceleration time as a friction of the natural frequency  $\omega_0$  [1/s]
2. Minimum acceleration/deceleration time defined by the installed pump pressure.
3. Minimum acceleration/deceleration time limited by the natural hydraulic operating time of the proportional device.

**Notes**

Notes

Chapter F

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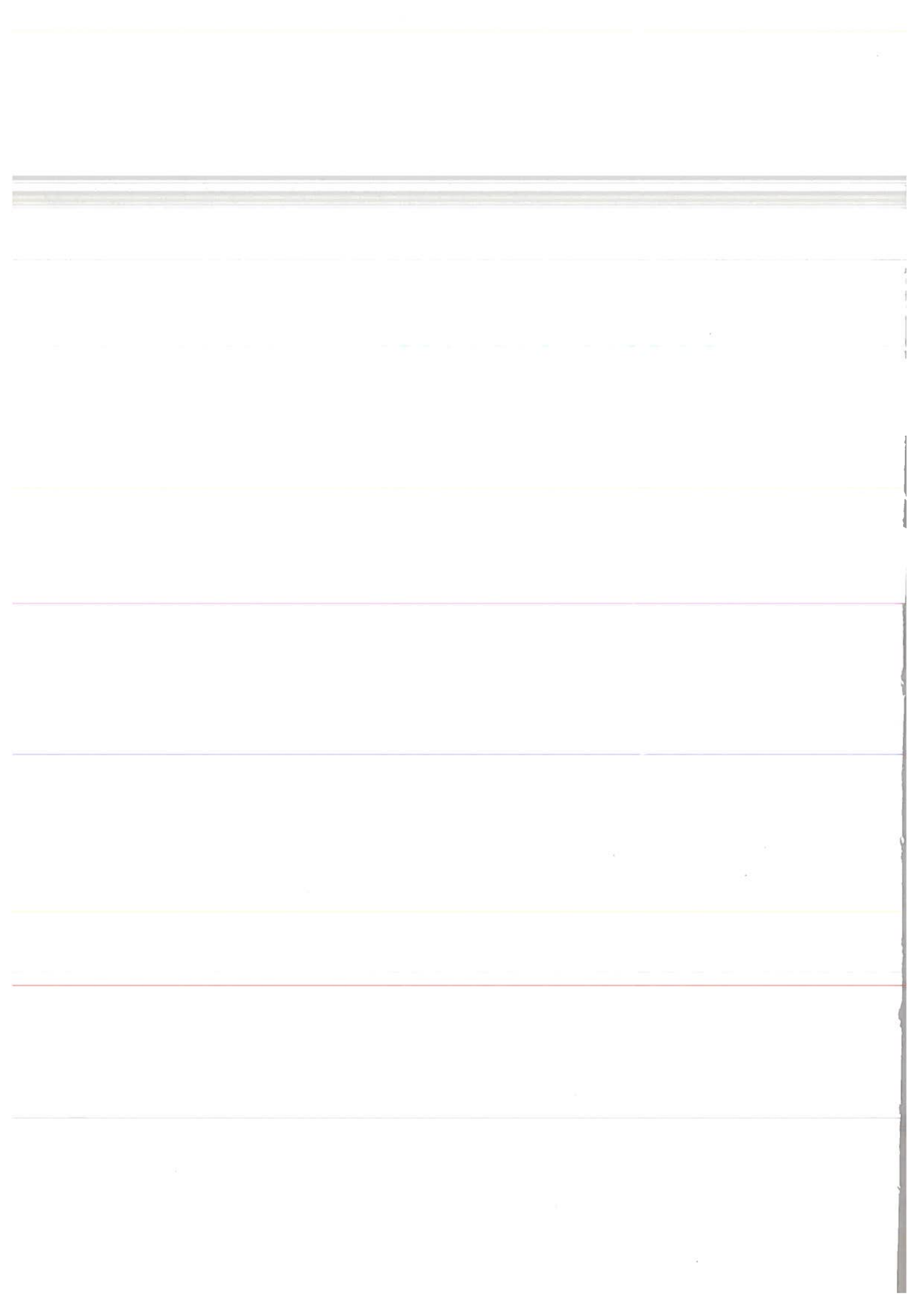
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## **Introduction to Servo Valve Technology**

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Dieter Kretz



## Development of Electrohydraulic Servo Valves

The fundamental development of servo hydraulics systems stems from aviation applications. Electrohydraulic servo valves were designed and constructed to precisely control flying objects with extremely small electrical input signals. The changeover from electrical or electronic control systems to electrohydraulic open loop and closed loop controls was particularly due to the high flying speeds and the resulting high positioning velocities and forces.

It was therefore necessary for the positioning device to meet demanding requirements with regard to speed, precision and power density.

In the course of time, industry has also implemented this technology, modified to suit the accuracy required in industrial applications so that it was possible to offer the devices at prices acceptable to industry.

### Definition of Servo Hydraulics

The term "servo hydraulics" is widely used in technical jargon. Nevertheless, a great many differing opinions still exist regarding to its true meaning.

For instance, it could be expressed as being "closed loop electrohydraulic control".

This definition could include all applications in which hydraulic devices operate in closed loop control circuits.

Operating in closed loop control circuits means the operating status is constantly monitored by means of measurement while deviations from the required operating status are automatically corrected.

The controlled variables are mainly mechanical variables

such as	- distance	or	angle of rotation
	- velocity		rotary speed
	- force		torque

or hydraulic variables

such as	- volumetric flow
	- pressure

To be able to control the above specified variables, corresponding measuring devices are necessary for determining the actual value.

To facilitate assessment of closed loop electrohydraulic control circuits or to define their power limits, it is necessary for the user to examine the areas:

Closed loop technology

Electronics

Hydraulics and

Measurement technology.

### Servo Hydraulics as a System

It will be apparent that servo hydraulics is pure control technology.

All elements involved in the closed loop control must be taken into consideration.

The result depends to a great extent on the intensive cooperation of all persons working on a project, since good cooperation at the earliest possible stage is the best prerequisite for obtaining optimum results.

Compromise solutions are often the result when cooperation begins at a point in time at which the essential features of a project have already been irreversibly defined.

Servo hydraulics therefore not only refers to individual hydraulic components but much more to the interaction of applied closed loop control, hydraulics for power transmission and electronics for data processing.

### Difference Between a Control Chain and a Closed Loop Control Circuit

#### Control Chain (Open loop control)

If the switch "a" is connected, the proportional amplifier "b" actuates the proportional directional valve corresponding to the set signal value. The proportional valve opens, thereby enabling flow.

The piston rod of the cylinder Z moves.

If it is now necessary to stop the piston of the cylinder at a defined, reproducible point when the switch is opened, then this is only possible to a limited extent.

The reasons are:

- The switching characteristic of the proportional valve varies with the oil viscosity.
- The pressure drop at the valve varies due to viscosity-dependent losses in the piping system.
- Various flow rates result from differing  $\Delta p$  and produce various positioning velocities of the cylinder.
- The deceleration distance changes dependent on the mass moved and the positioning velocity.

All these "interfering variables" effect the result obtained by a control chain.

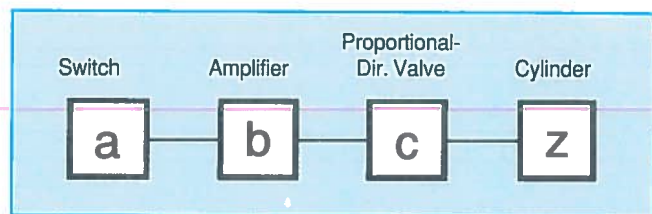


Fig. 1 Block diagram of a control chain

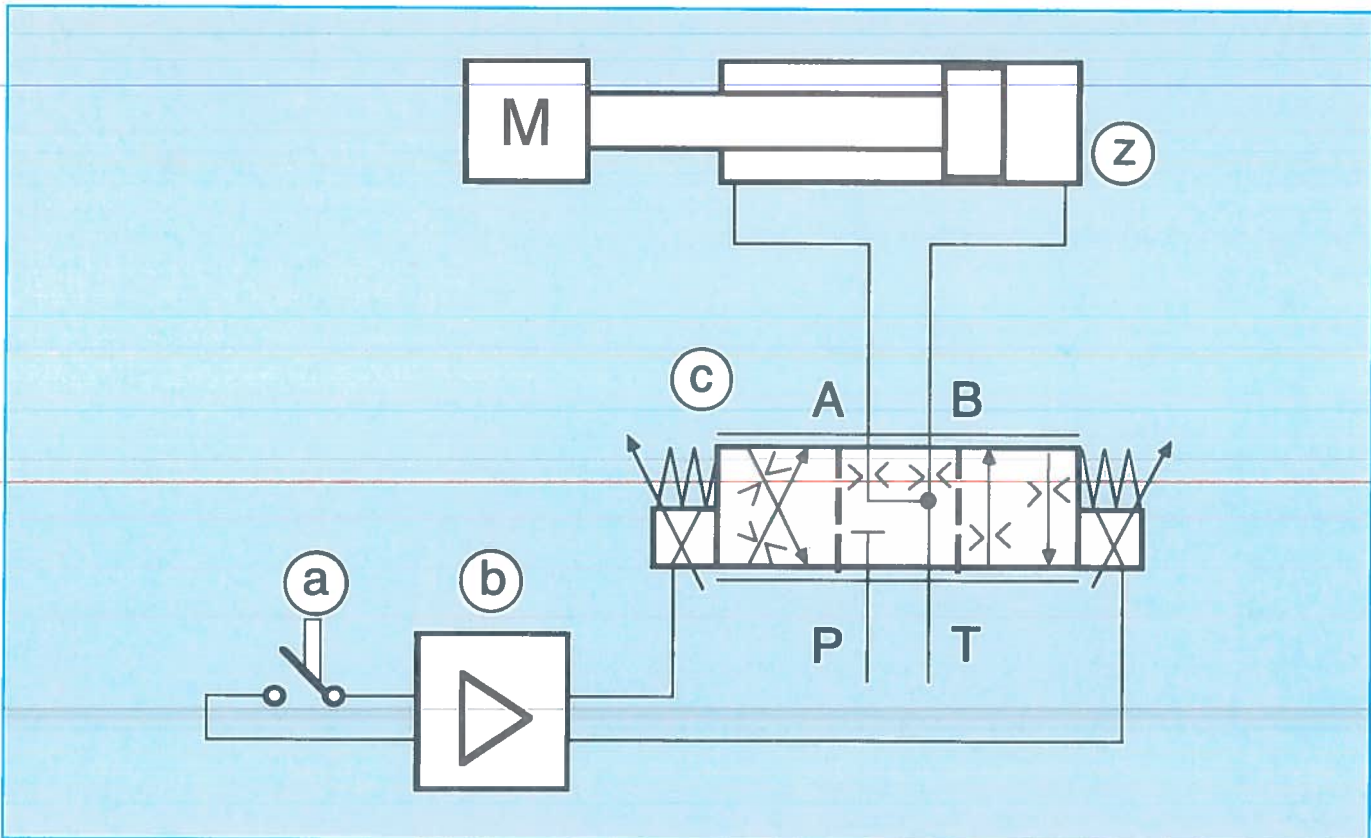


Fig. 2 Open loop control with proportional directional valve

### Closed Loop Control Circuit

With potentiometer P1, a signal value voltage is pre-selected which corresponds to a certain position of the piston. The actual position of the piston, i.e. the actual value, is also provided in the form of a voltage by potentiometer P2. These two voltages are subtracted from each other at the input of the amplifier V, i.e. the signal/feedback difference - the closed loop error - is produced. The error is amplified in the amplifier V and is now able to energize the coil of the servo valve SV. As a result, the servo valve opens and the piston moves. At the same time, the position of the potentiometer P2 also changes, the actual value voltage approaches the signal value voltage ever more closely, until it is finally equal to P1 when the required position is reached. The error constantly becomes smaller

during this procedure and despite amplification, less and less current is available to the coil of the servo valve. This means the servo valve closes gradually and therefore decelerates the piston. When the required position is reached, the error is zero and the servo valve closed

It can be seen that the interfering variables described in the open control chain no longer or scarcely effect the result in the closed loop control circuit. This is an important feature of closed loop technology and therefore also of servo hydraulics.

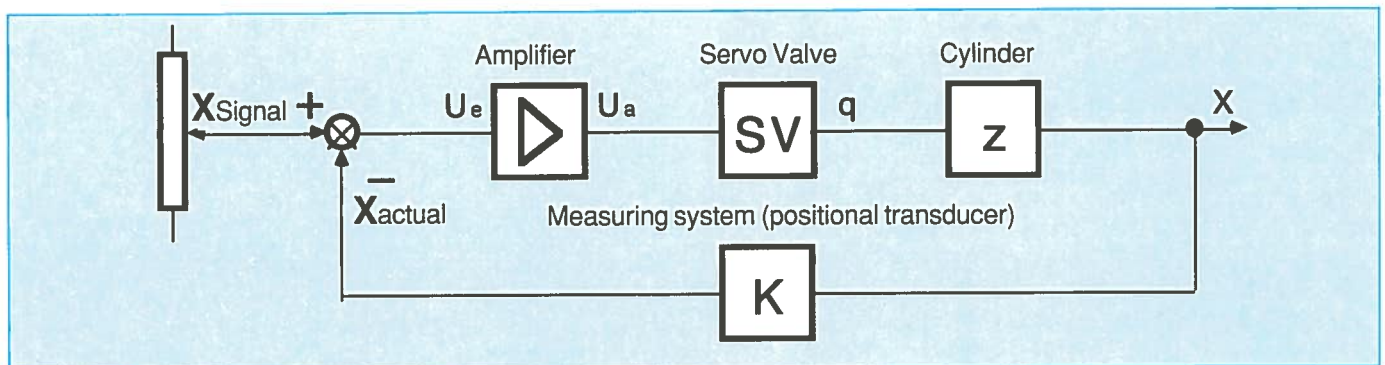


Fig. 3 Simplified block diagram of a closed loop control circuit

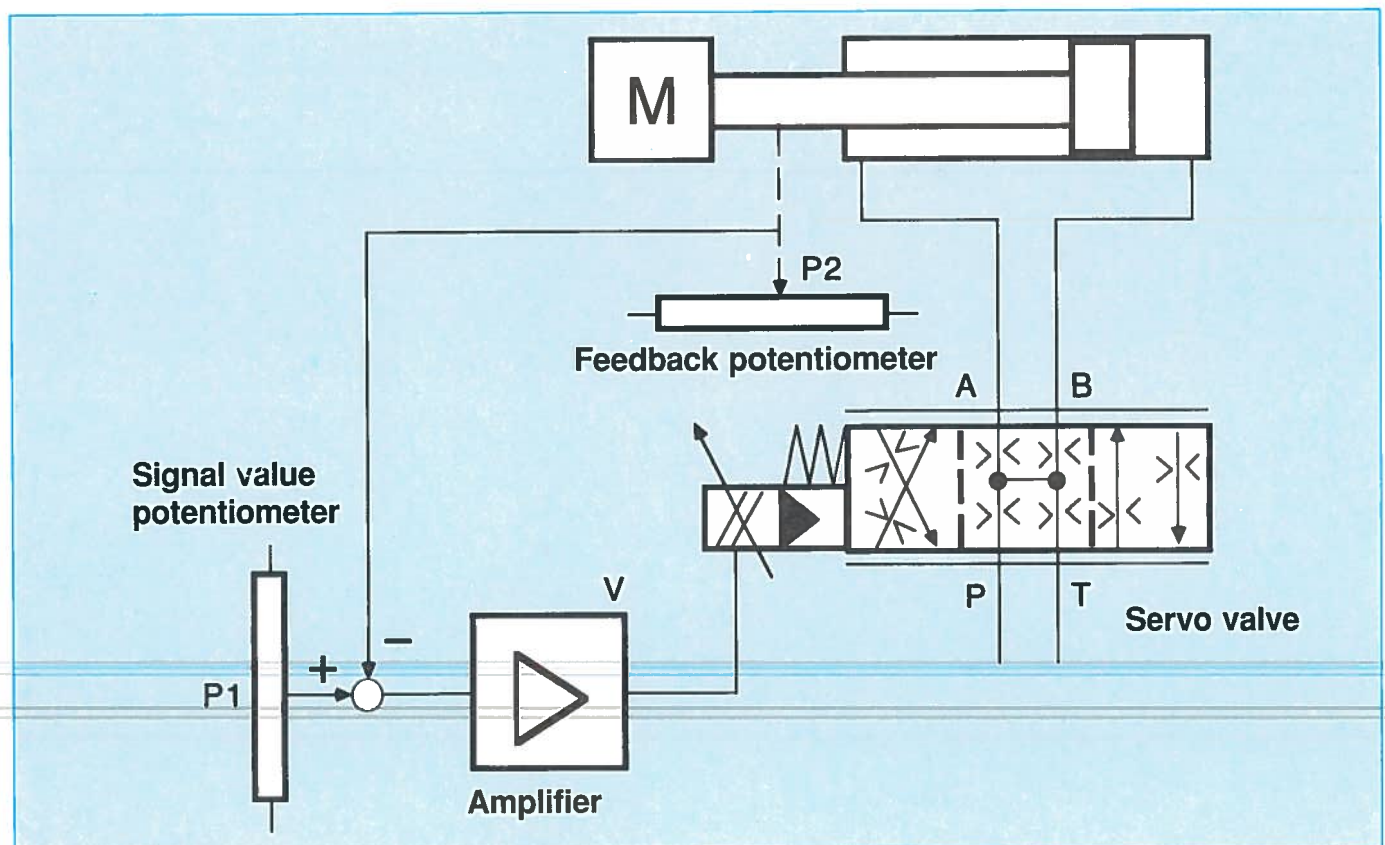


Fig. 4 Closed loop control circuit with servo valve

## Definition, Data and their Significance in Applications

The description of servo valves includes a great number of terms and definitions which should firstly be defined and their meaning explained.

### 1. Static Data

#### 1.1 Nominal Flow

The nominal flow of servo valves is normally referred to a total pressure drop of 70 bar.

This does not mean, however, that operation is limited to this 70 bar pressure drop. Any operating point (flow) can be defined.

$$Q = Q_{\text{nominal}} \cdot \sqrt{\Delta p / \Delta p_{\text{nominal}}}$$

$Q_{\text{nominal}}$  = Nominal flow at nominal pressure drop  $\Delta p_{\text{nominal}}$

The nominal flow is always referred to complete spool movement of the servo valve. In the case of partial spool movement the flow varies proportional to the movement ratio.

#### 1.2 Flow Curve

The flow curve shows the relationship between the valve flow and the electrical input signal.

A, B = Characteristic operating points  
 A = Operating point about zero point  
 B = Operating point when open

Refer to 1.3 (Page F6) for significance of operating point in control systems.

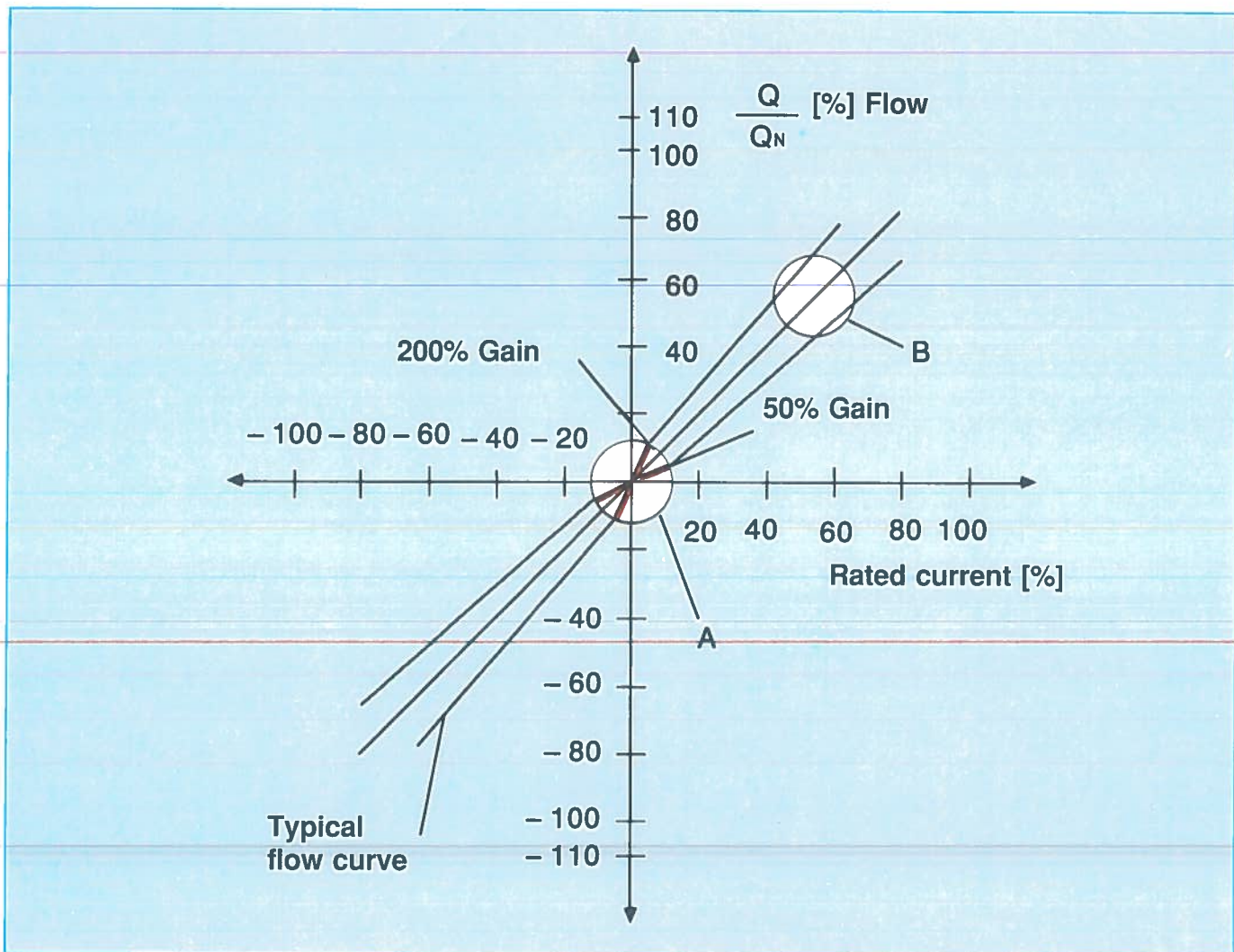
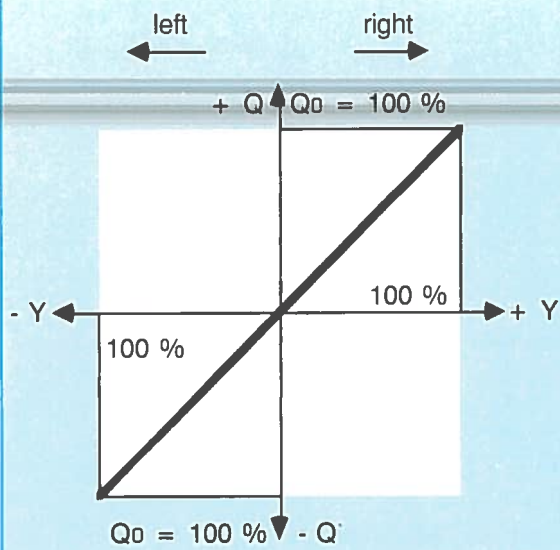
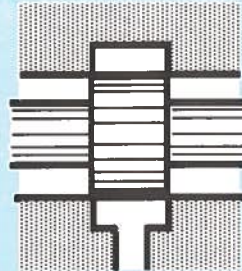


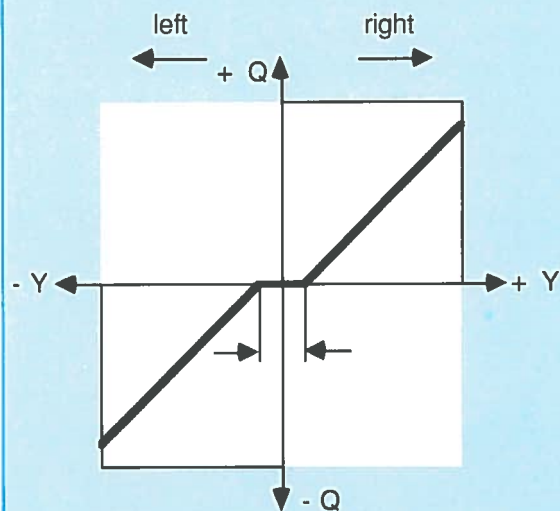
Fig. 5 Flow curve

**"Zero" Overlap**

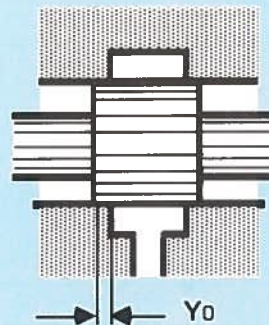
There is no volumetric flow in control spool position  $Y=0$ .  
There is constant volumetric flow via the control land in position  $|Y| > 0$ .



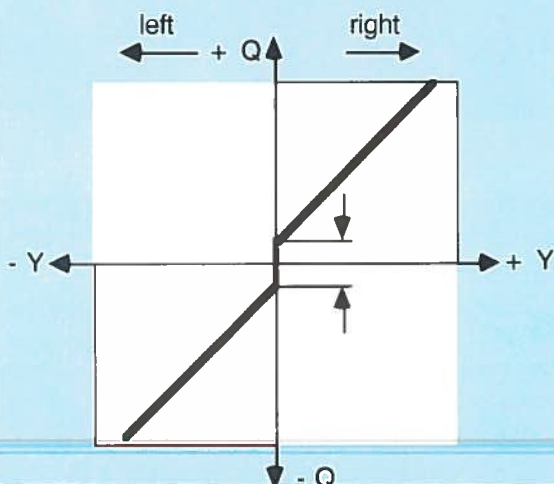
$$Y_0 = 0$$

**Positive Overlap**

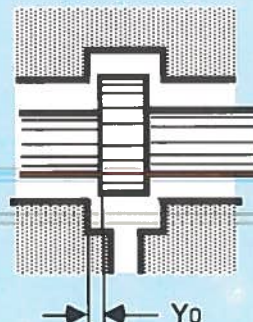
Control openings remain closed in the area  $|Y| \leq Y_0$ .  
There is constant volumetric flow via a control land in position  $|Y| > Y_0$ .



$$Y_0$$

**Negative Overlap**

There is constant volumetric flow via both control lands in the area  $Y = Y_0$ .  
There is volumetric flow via only one control land in position  $|Y| \geq Y_0$ .



$$Y_0$$

Fig. 6 Flow curves for various degrees of overlap in the neutral point (Point A)

## 1.3 Assignment of Overlap to Control Function

### - Position and pressure control

In closed loop position and pressure control, the valve operates in the working point "A", i.e. about the neutral point.

Zero overlap or a negative overlap must be selected for this application. Positive overlap cannot be used here since signals are not transferred within the overlap area or signals are transferred only in distorted form immediately outside the overlap area. Stable closed loop control is therefore not possible.

### - Closed loop velocity or flow control

In a closed loop velocity control, the valve operates in the working point "B". In this case, a positive overlap in the neutral point can be used.

### - Isolating function with positive overlap

A positive overlap does not represent reliable cut-off. The overlap is normally small to ensure sufficient flow is achieved with the remaining stroke. Due to the neutral point being shifted as the result of pressure and temperature fluctuations or a jet being blocked on one side, the flow is enabled in one direction and the drive moves.

## 1.4 Flow Amplification

Amplification is normally specified as the relationship between the output signal and input signal. Flow amplification is therefore defined as

$$V_q = q/U_E \quad [\text{L/min/Volt}]$$

This definition represents the average increase in the flow curve. The gradient of this curve depends on the system pressure.

Due to production tolerances, various degrees of amplification occur particularly about the neutral point (refer to flow curve, Fig. 6). As a result, it may be necessary to readjust the controller when replacing the valves.

## 1.5 Response Sensitivity "E" and Reversal Range "S"

### - Sensitivity of response

The term sensitivity of response refers to the change of the electrical input signal which is necessary to produce a measurable change in flow when the signal is changed from a stopping point in the same direction as the stopping point was approached. The value is specified as a percentage of the rated current.

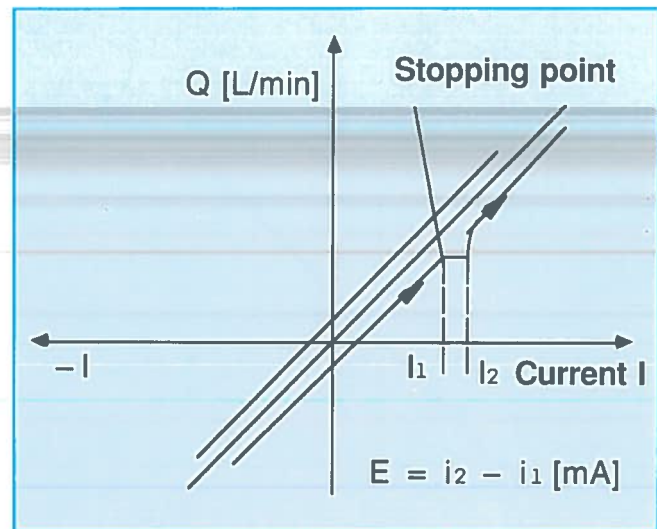


Fig. 7 Sensitivity of response

### - Reversal range

The reversal range is the change in the electrical input signal necessary to produce a change in flow when the signal is changed from a stopping point in the opposite direction as the stopping point was approached. Specified in percent of rated current.

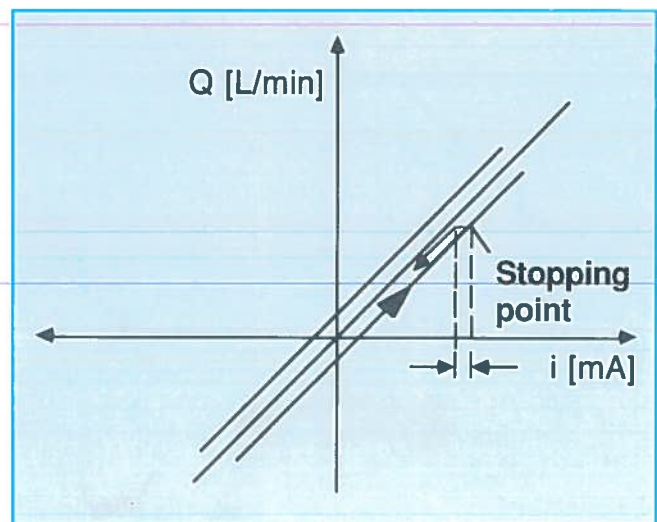


Fig. 8 Reversal range

The response sensitivity and the reversal range therefore represent dead zones which influence the accuracy of the closed loop control circuit.

If the servo valve is to perform a correction, then it requires an input signal which, depending on the direction of the correction, must be greater than the sensitivity of response or the reversal range.

An input signal is produced by a closed loop error, i.e. the difference between signal and feedback value. Neglecting the pressure ratios, this means that the control range in the case of closed loop flow control and the possible positioning accuracy in closed loop positional control are directly influenced by the servo valve.

### 1.6 Pressure-Signal Function

A corresponding force is necessary in order to correct a drive. For this reason, the progression of the output pressure with respect to the input signal is of considerable significance. This progression is shown in the characteristic pressure curve.

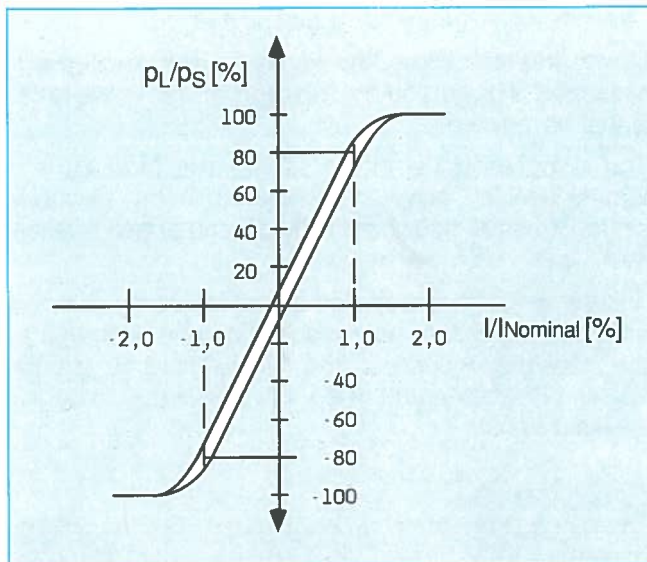


Fig. 9 Pressure-signal function

### 1.7 Pressure Amplification

The relationship between the output pressure and input signal is referred to as pressure amplification.

$$V_p = p_L / U_E \quad [\text{bar/Volt}]$$

The pressure curve shows how far the servo valve must open in order to pass the pressure required for correction.

The opening of the valve is in turn controlled by a closed loop control circuit so that the pressure amplification directly effects the closed loop control accuracy. Correspondingly, the pressure amplification should be as large as possible.

In the case of the pressure curve illustrated, 80 % of the system pressure is available for correction of the closed loop error at 1 % of the rated current.

### 1.8 Flow/Load Function

A servo hydraulic drive generally consists of a servo valve and a cylinder or motor as a load. Movements are influenced by throttling the supply oil flow.

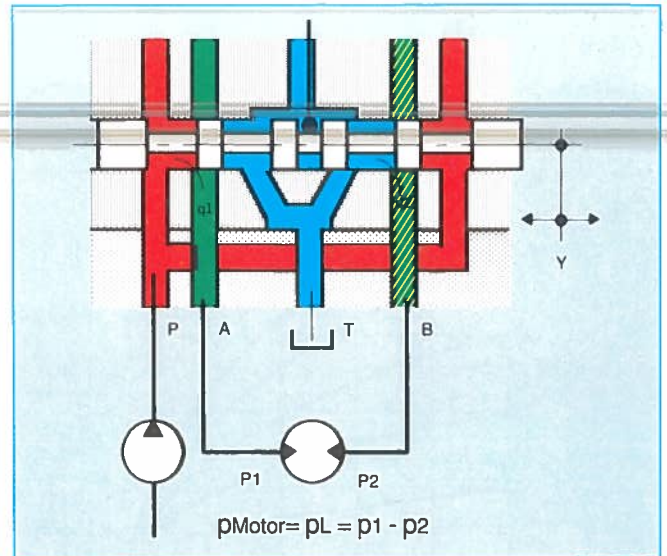


Fig. 10 Four edge throttle control

Assuming ideal conditions, the oil flow through a throttling point is obtained in accordance with the following equation

$$Q = Y \cdot K \cdot \sqrt{\Delta p}$$

In this equation, Q is the oil flow, Y the degree of spool movement (= percentage movement, see fig. 11) and K a constant which takes into consideration the geometry of the control opening, the density of the oil etc. and  $\Delta p$  is the pressure drop at the control land.

Depending on the load, the motor connected in the example requires a load pres.  $p_L$ . If  $p_S$  is the system pressure, then the following pressure drop remains

$$\Delta p = p_S - p_L$$

$$Q = Y \cdot K \cdot \sqrt{p_S - p_L}$$

When the motor is not subject to load, i.e.  $p_L = 0$ , the entire system pressure is available as  $\Delta p$ . The maximum oil flow is provided. If the motor is blocked, the entire system pressure is applied at the motor the oil flow is then zero.

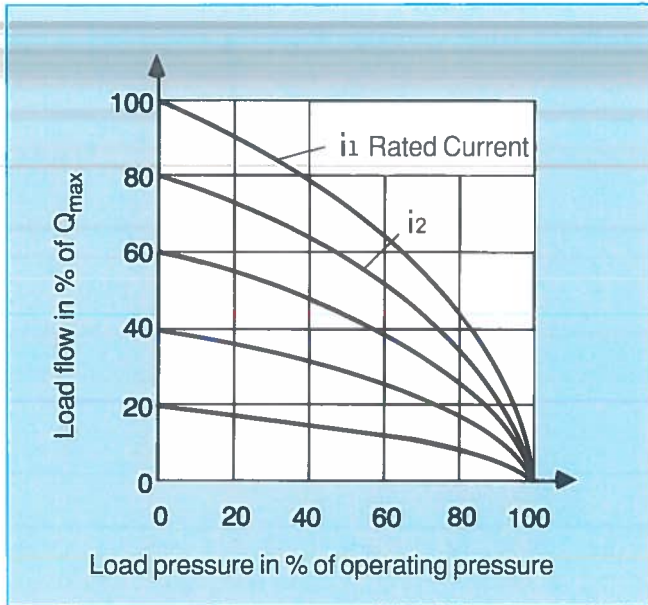


Fig. 11 Flow/load function,  $i_1 = i_{max}$

## 2. Dynamic Data

The natural frequency of a drive and the resulting total gain possible are decisive for the closed loop control accuracy of the relevant drive. The natural frequency of the drive is primarily defined by the dynamics of the servo valve.

Simply stating the positioning time is not sufficient in this case to describe the dynamic characteristics. The most common method of examining the dynamic characteristic is the frequency response method.

During this procedure, the servo valve is excited with sinusoidal signals and the reaction of the valve to the signals is registered.

The response signal of the servo valve (flow  $Q$ ) is in turn sinusoidal, however, compared to the excitation signal, it has a modified amplitude and phase relationship.

Initially, the signal is applied at low frequency and then gradually increased. As a result, it can be seen that as the frequency increases, the initial amplitude decreases and the movement of the valve always lags behind the input signal.

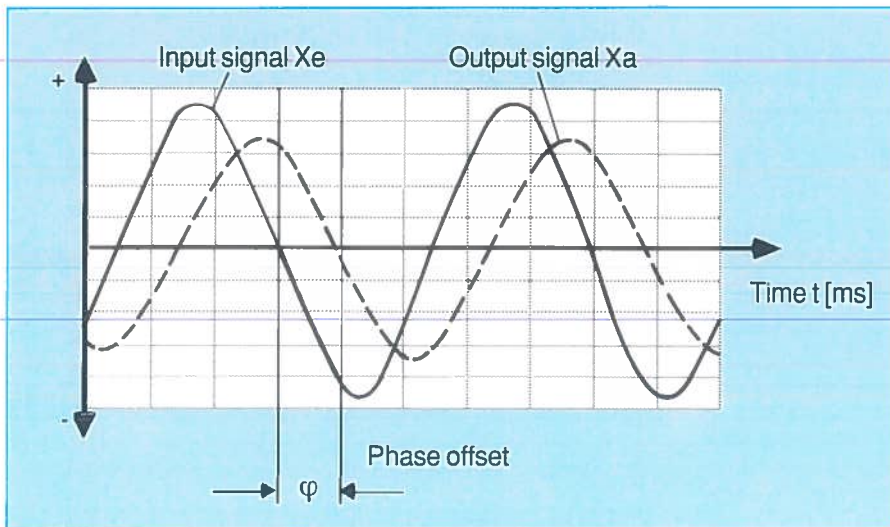
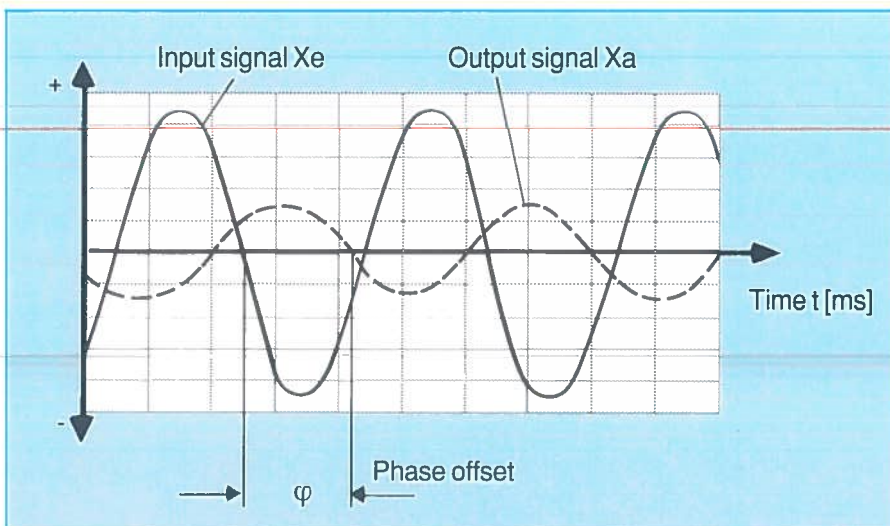


Fig. 12a and 12b Frequency response curve



## 2.1 The Bode Diagram

This relationship is illustrated by the Bode diagram.

Here, the relevant relationship of the output amplitude with respect to the input amplitude  $X_a/X_e$  is marked over the excitation frequency to obtain the "amplitude frequency relationship". Furthermore, the phase offset of the output signal is marked in this diagram with respect to the input signal over the frequency to obtain the "phase frequency characteristic". Both curves together form the Bode diagram.

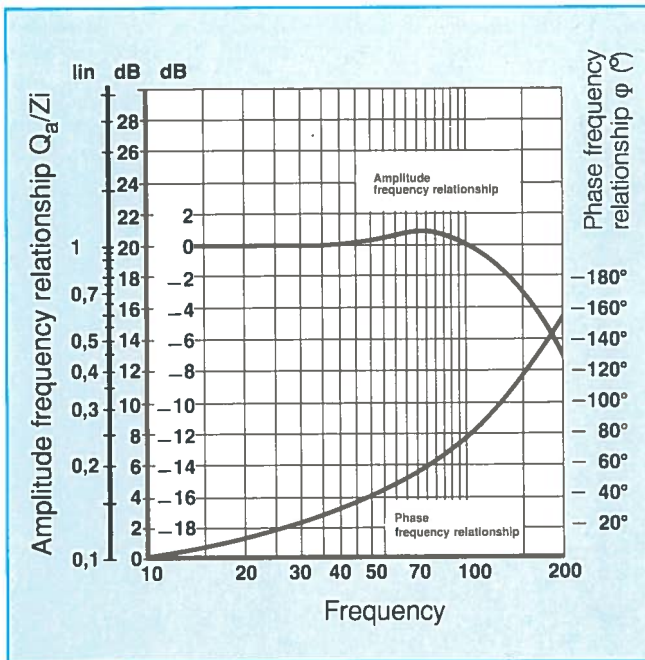


Fig. 13 Bode diagram

The amplitude frequency relationship is normally specified in dB (decibel).

Therefore

$$\text{amplitude ratio in dB} = 20 \cdot \log X_a/X_e$$

or

$$X_a/X_e = 10^{(\text{dB}/20)}$$

To facilitate pure qualitative description of the frequency response, the characteristic values for frequency have been defined as -3 dB and -90°.

The frequency is defined as  $f_{-3 \text{ dB}}$  at which the output signal  $Q$  of the valve has dropped by -3 dB with respect to the input signal, corresponding to the ratio  $X_a/X_e = 0.707$ . This characteristic value describes a point on the amplitude frequency relationship curve.

The  $f_{90^\circ}$  frequency describes the point on the phase frequency characteristic curve at which the output signal lags behind the input signal by 90°.

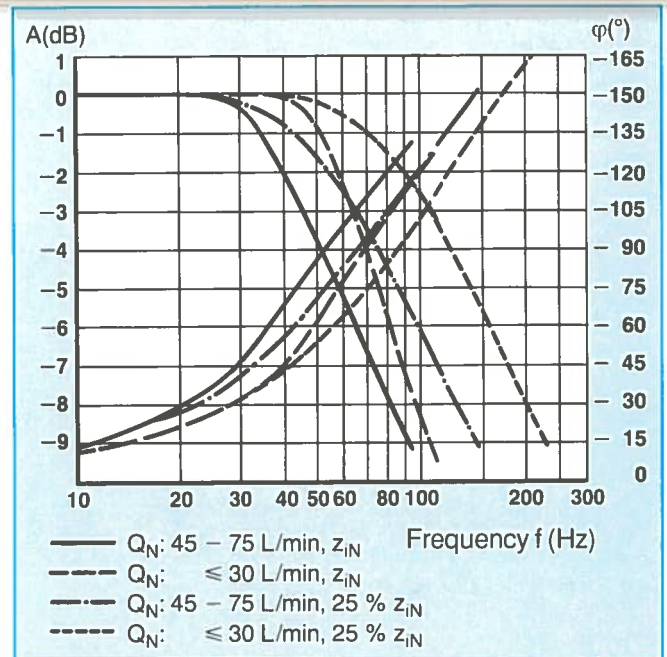


Fig. 14 Frequency response of a size 10 servo valve with mechanical feedback

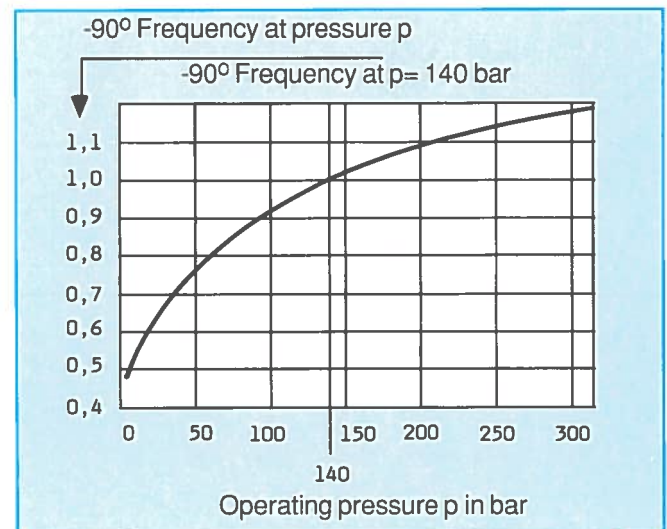


Fig. 15 Function of operating pressure

The dynamic characteristic of the servo valve is considerably influenced by the system pressure  $p_S$  and the level of the input signal  $I/I_{\text{nominal}}$ .

The data can be obtained directly from the frequency response for an operating pressure of 140 bar.

For other operating pressures, the frequency obtained from the phase frequency characteristic for the -90° point must be multiplied by the factor obtained from Fig. 15.

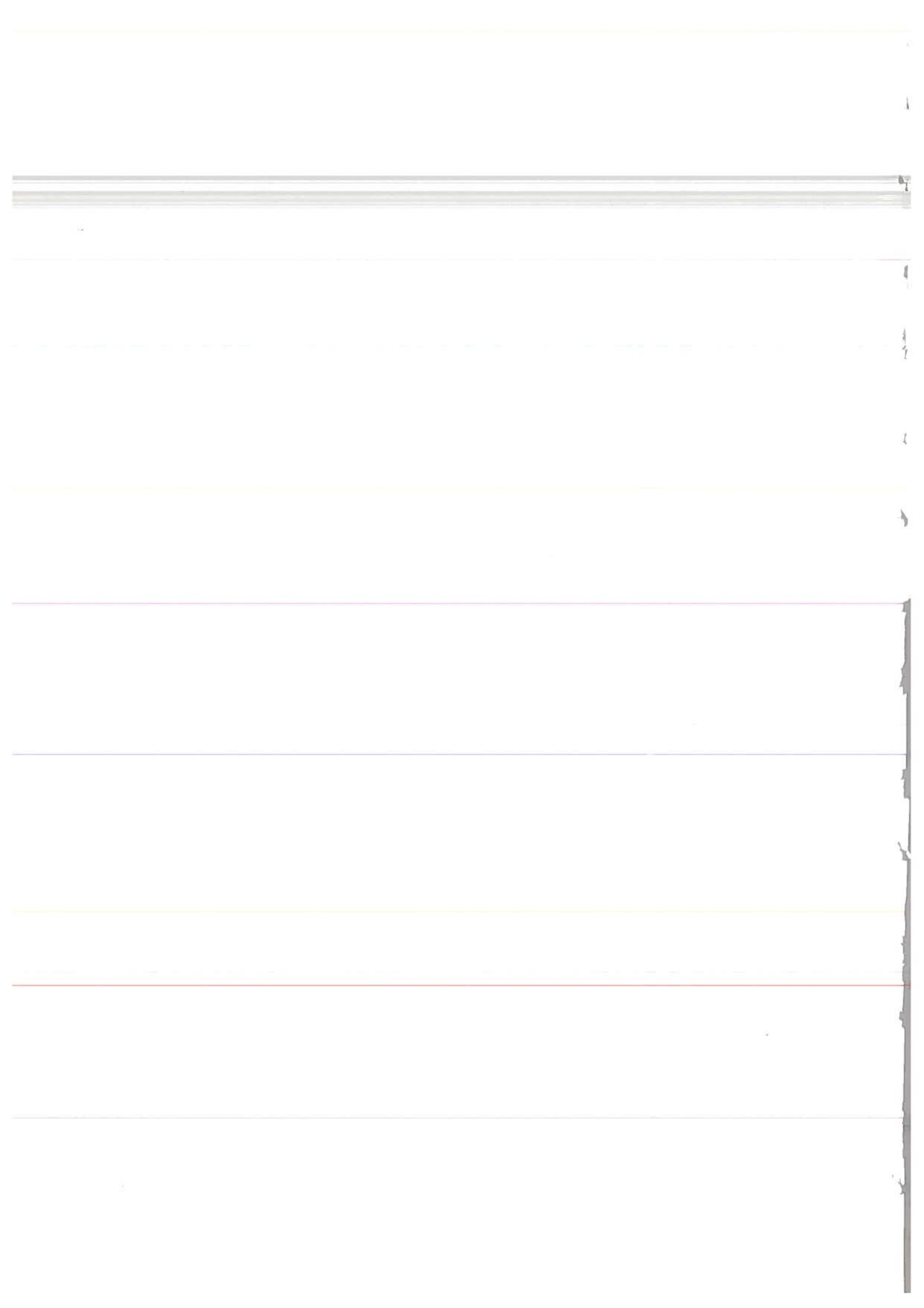
Notes

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## **Servo Valves, Device Technology**

Friedel Liedhegener



## General Information

Rexroth servo valves have been developed as industrial valves and comply with the requirements of industry with regard to reliability, interchangeability and easy servicing. They are of modular design.

This includes,

- the constant use of standardized mounting pattern to DIN 24 340 for all sizes,
- interchangeability of torque motors or pilot stages,
- externally adjustable
- interchangeable filter element in the pilot stage

Rexroth servo valves represent a further compact module within the sales range of Mannesmann Rexroth.

The term "servo" is used to describe a diverse variety of functions. Expressed very generally, this term refers

to the function in which a small input signal produces a large output signal (amplifier).

The best known example is probably the servo steering system (power steering) in a motor vehicle, where the steering wheel moved with little effort applies a great force on the wheels.

The procedure is similar in servo hydraulics.

A low power input signal, e.g. 0.08 Watt can actuate and control high power ratings of more than 100 kW.

The servo valve in the form of an electrically actuated hydraulic amplifier is primarily used in closed loop control circuits, i.e. not only is an electrical input signal converted into a corresponding oil flow, but rather the deviations from the preset speed or position are measured electrically and fed to the servo valve for correction.

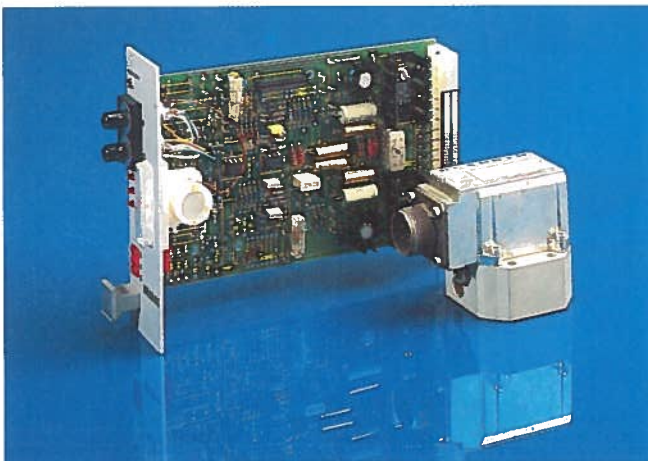


Fig. 1 Single-stage servo pressure valve  
Type 4 DS 1 EO 2,  
1st stage of the servo valve - modular system

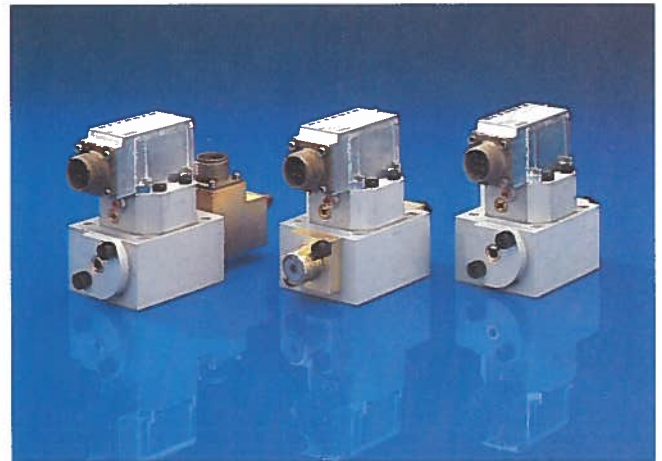


Fig. 2 Size 10 servo directional valves with mechanical feedback, Type 4 WS 2 EM 10 (right), with electrical feedback, Type 4 WS 2 EE 10 (left) and barometric feedback, Type 4 WS 2 EB 10 (centre)

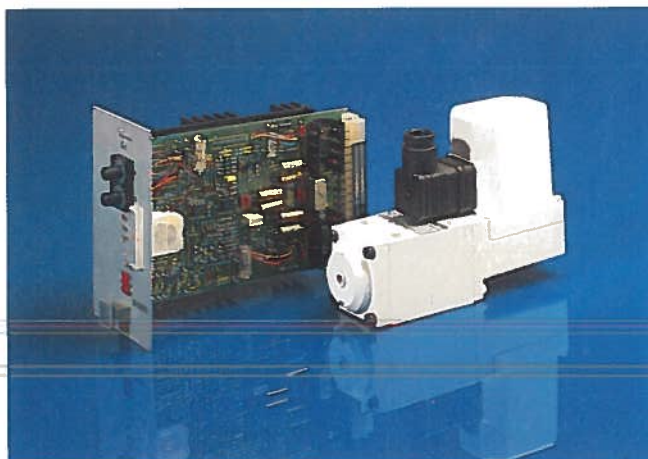


Fig. 3 Single-stage control valve (servo valve)  
Type 4 WS 1 EO 6

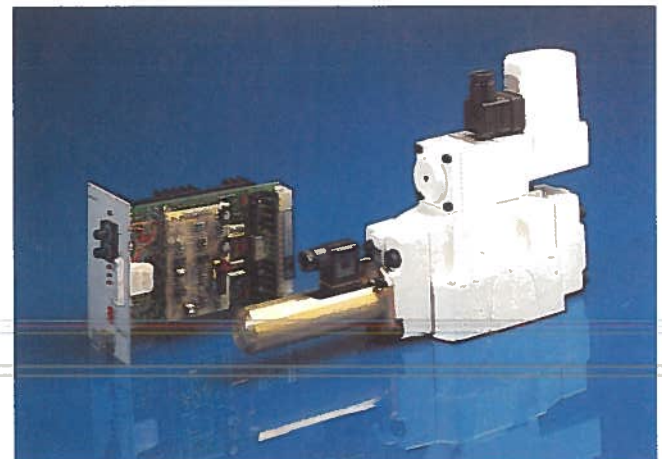


Fig. 4 2-stage proportional directional valve,  
Type 4 WRV, the control valve (Fig. 3) is used as the  
first stage

## Torque Motor

The torque motor converts a small current signal into a proportional, mechanical movement. In the case of Rexroth servo valves, the motor is a self-contained unit, it is mounted and tested separately and is interchangeable. Features which facilitate servicing and repair.

The "dry torque motor" is hermetically sealed from the hydraulic section and designed as described in the following:

An armature of magnetically "soft" material is mounted on a thin walled flexible metal tube, which also acts as centering spring, a seal for the pressure medium, and carries the so-called flapper plate. Thus, physically, the flapper plate belongs to the torque motor, but functionally forms part of the hydraulic amplifier.

Our torque motor represents a motor system excited by a permanent magnet. With the aid of adjustable "pole screws", the gap between the armature and pole screw can be adjusted thereby optimizing the motor characteristics.

The two coils arranged about the armature magnetize the armature. As a result, a bending moment (torque) is exerted on the "torque" tube, which also acts as a centering spring.

The torque is proportional to the level of the pilot current and equal to zero when the pilot current is switched off ( $I = 0$ ). As a result, the tube (return spring) returns the armature and therefore also the flapper jet to the mid-position.

The torque transmission of this type of torque motor from the armature to the flapper jet has clear advantages such as:

- Freedom of friction
- Low hysteresis
- Sealing pressure medium/torque motor
- No magnetic field in pressure medium

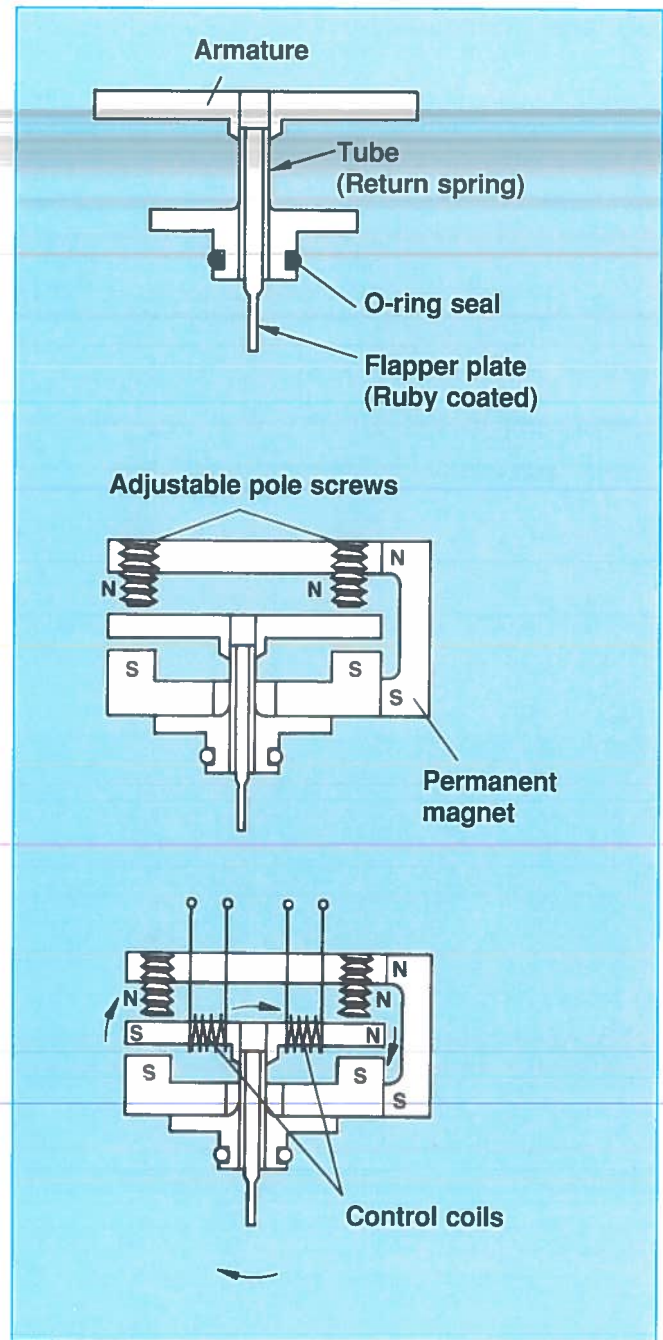


Fig. 5 Design of torque motor

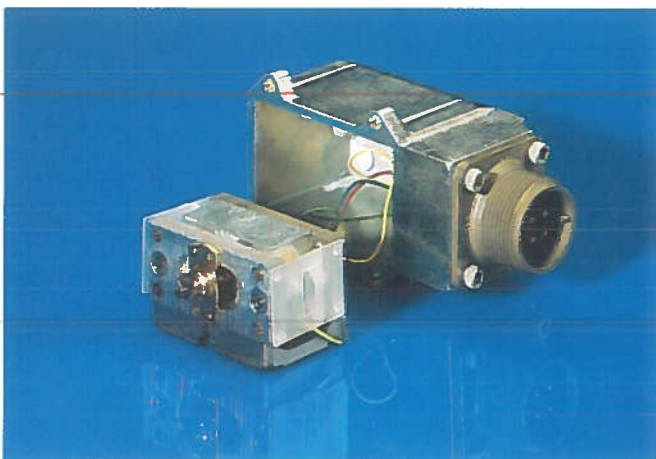


Fig. 6 Torque motor without feedback

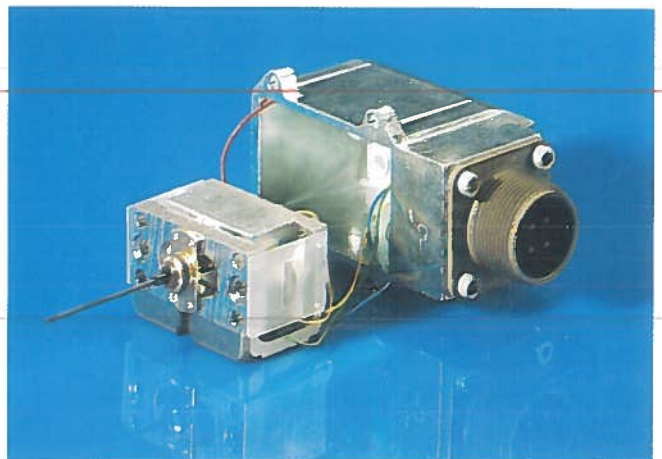


Fig. 7 Torque motor with mechanical feedback

## 1st Stage

Type 4 DS 1 EM 2 valves are single-stage servo pressure valves and are used for the pilot control of multi-stage servo valves.

They basically consist of

- the torque motor (1) excited by a permanent magnet
- the hydraulic amplifier (2) designed as a flapper jet valve

### Torque Motor

The torque motor is a motor which is excited by a permanent magnet and hermetically sealed from the hydraulic section.

An armature (3) made of magnetically soft material is secured in a spring arrangement to a thin-walled, flexible tube (4). This tube also serves to carry the flapper jet (5) and seals the torque motor (1) from the hydraulics. The distances between the armature (3) and the upper pole plate (8) can be adjusted with the pole screw (6).

The magnetic flux is equal in the four gaps (9) when the distances are equal and no electrical signal is applied. The armature (3) is offset when an electrical control signal is applied to the coils (10). The flapper jet (5) is also displaced together with the armature (3).

The torque produced in the armature (3) by the pilot current behaves proportionally with respect to the electrical input signal and is equal to zero when the pilot current is switched off ( $I = 0$ ). As a result, the armature and the flapper jet are held in the centre position by the tube (4).

### Hydraulic Amplifier

The offset of the flapper jet is converted into a hydraulic variable in the hydraulic amplifier (2). In this case, the flapper jet system is used as a hydraulic amplifier (Fig. 8).

The system consists of two fixed jets  $D_1$  and 2 control jets  $D_2$ . The pilot pressure  $p$  applied at both sides is reduced via the jets  $D_1$  and  $D_2$ . If the jet openings are of equal size, the same pressure drop is also obtained via the jets (e.g.  $p = 100$  bar,  $A_{St}/B_{St} = 50$  bar,  $T = 0$ ).

The distances to the control jets also changes with the offset of the flapper jet, in example offset to left:

The distance of the flapper jet at  $D_2$  left becomes smaller and at  $D_2$  right greater. The pressure at  $A_{St}$  and  $B_{St}$  change conversely. Pressure  $A_{St}$  increases, whilst pressure  $B_{St}$  decreases. The pressure difference  $A_{St} - B_{St}$  is used as the effective signal.

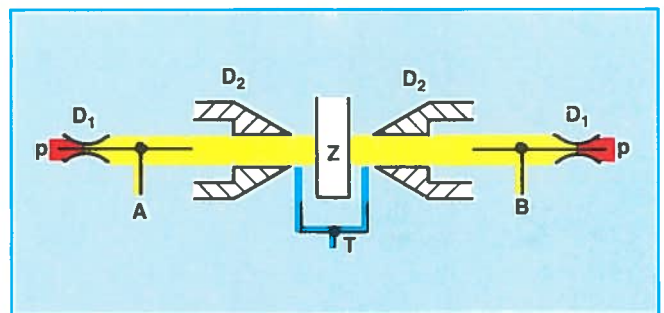


Fig. 8 Principle of flapper jet system

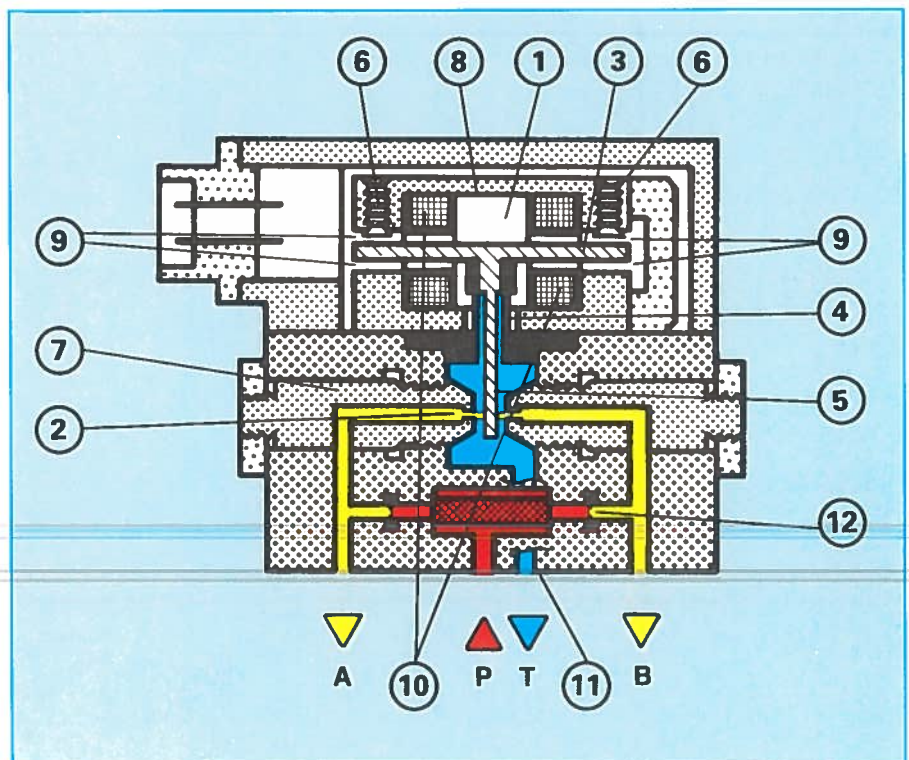


Fig. 9 Schematic diagram of 1st stage

The diagram (Fig. 10) shows the change in pressure dependent on the offset.

The control is arranged, so that a linear curve is obtained (pressure difference between the ports  $A_{St}$  and  $B_{St}$ ).

The pilot oil is fed from port B via a protective filter (11) to the fixed jets (12) and further to the control jets (7).

The pressure  $A_{St}$  and  $B_{St}$  is tapped off between the fixed jets and control jets.

This pressure difference which is proportional to the electrical input signal is now further applied to the control spool of a second stage.

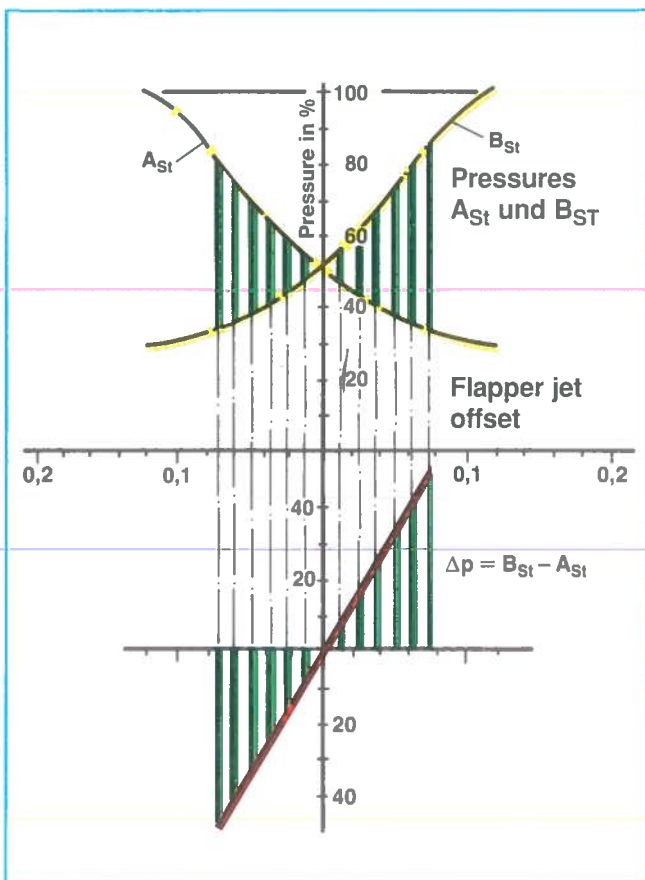


Fig. 10 Change in pressure as a function of flapper jet offset

## 2-Stage with Mechanical Feedback

The 2-stage servo directional valve basically consist of:

- the 1st stage
- the mechanical feedback (3) as a connecting element between the 1st stage and 2nd stage;
- the 2nd stage with the interchangeable control sleeve (4) and the control spool (5) coupled to the mechanical feedback (3).

### 2nd Stage

The control spool (5) is linked via the mechanical feedback (3) almost free of play to the torque motor (1) of the 1st stage.

The type of feedback used in this case functions dependent on the torque balance between the torque motor (1) and feedback spring (3).

This means, in the case of unequal torque, caused by a change in the electrical input signal, the flapper jet (6) is firstly moved out of the centre position between the control jets. As a result, a pressure difference is produced which acts on both ends of the control spool. The effect of the pressure difference changes the position of the control spool (5). As a result of this change in position of the control spool (5), the feedback spring (3) bends until the flapper jet is pulled back to the centre position to such an extent that the main spool stops and torque balance is established.

A spool stroke proportional to the input signal and therefore also the flow are therefore re-established.

With the aid of the two socket screws (8) which are located to the left and right in the valve covers (9), the position of the control sleeve (4) can be shifted with respect to the control spool (5) in order to adjust the hydraulic neutral point.

### Special Valve Features

The connection dimensions of the main stage (2nd. stage) of this valve correspond to DIN 24 340.

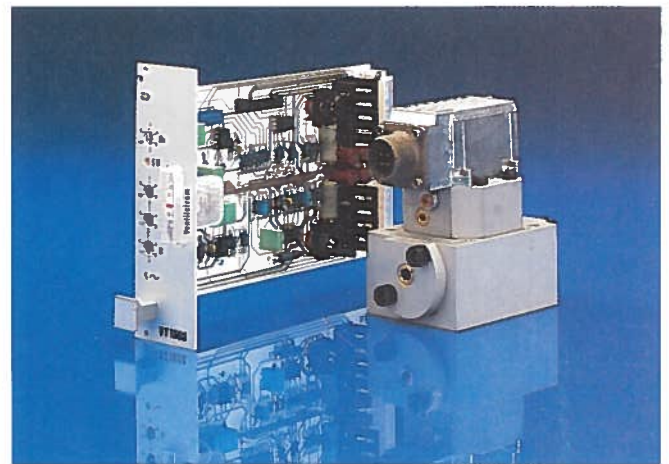


Fig. 11:  
2-stage servo directional valve Type 4 WS 2 EM 10

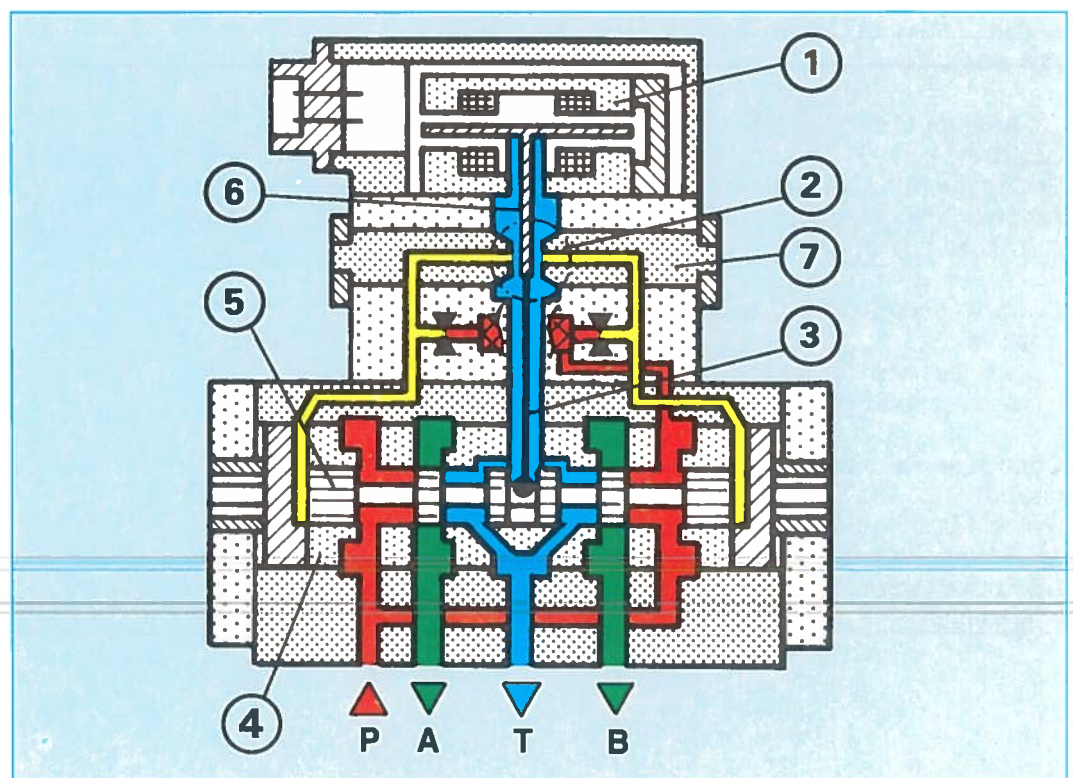


Fig. 12:  
2-stage servo directional valve with mechanical feedback,  
Type 4 WS 2 EM 10

## Flow Curves

Depending on the application for which the servo valve is used, in addition to the dynamic characteristics, two hydraulic characteristic values are of particular importance:

Flow amplification and spool overlap (defining pressure amplification)

### Flow Amplification (Fig. 13)

The control sleeve features rectangular control windows, which are opened by the main spool corresponding to the input signal. The width of these slots determines the flow amplification (flow per spool stroke). The flow rate is specified in l/min which flows at 70 bar valve pressure drop (i.e. 35 bar from P→A and 35 bar from P→T) and at 100 % input current. In the case of higher flow amplification, the flow curve dips as a result of housing saturation.

### Spool Overlap (Fig. 14)

The four control lands of the main spool are ground in symmetrically. Thus, one of four positive overlap or negative overlap sizes (in % of spool stroke) can be chosen. In the case of positive overlap, the curve is shallower in the centre area. In the case of negative overlap, the curve can be steeper in the vicinity of the centre (flow amplification up to 200 %). The zero position flow is higher, the pressure amplification lower.

Main applications:

#### Control spool overlap A (+0.5...1.5 %), positive

Suitable for velocity control loop.

Advantage: Lower zero position flow than in "D".

#### Control spool overlap B (-0.5...1.5 %), negative

Suitable for closed loop position and force control.

Advantage: Higher damping, however greater zero position flow than "D".

#### Control spool overlap C (+3...5 %), positive

Suitable for open loop controls and velocity control loops without zero position flow.

#### Control spool overlap D (+0...0.5 %), neutral

Suitable as universal overlap for closed loop speed, position and force control.

Advantage: Lower zero position flow, but lower damping than "B".

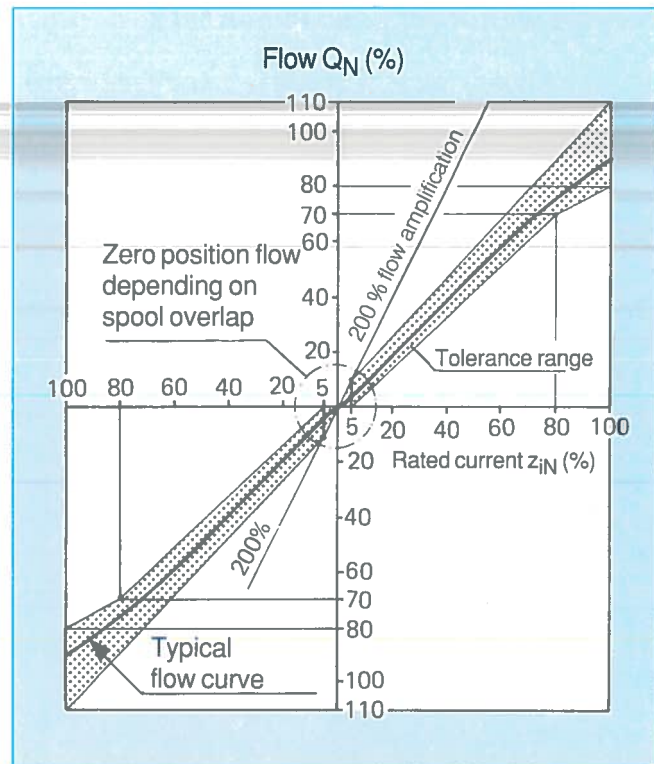


Fig. 13 Tolerance range of flow/signal function

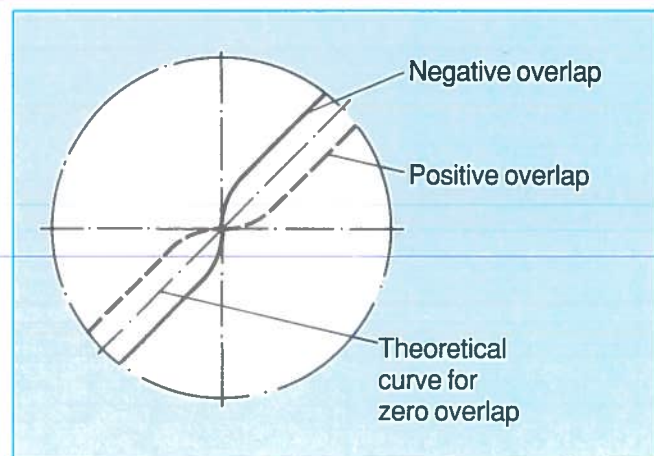


Fig. 14 Principle of spool overlap

## Dynamics of the Servo Directional Valve

The dynamic characteristics of the valve are indicated by the frequency curve. Control technicians have defined the frequency, at which the amplitude frequency relationship is -3 dB. -3 dB means that the drop in amplitude of the output variable is 30 % of the input variable.

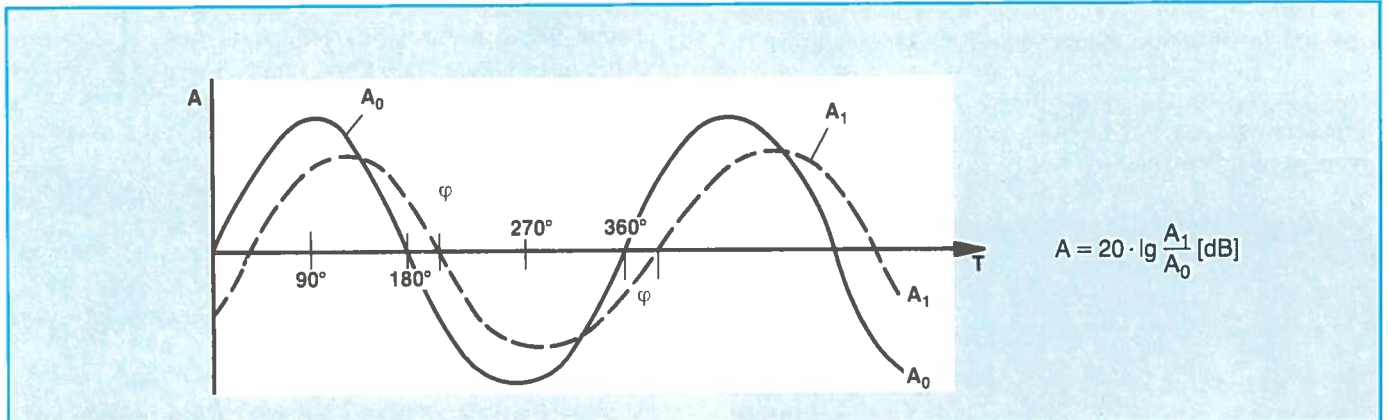


Fig. 15 Drop in amplitude and phase offset

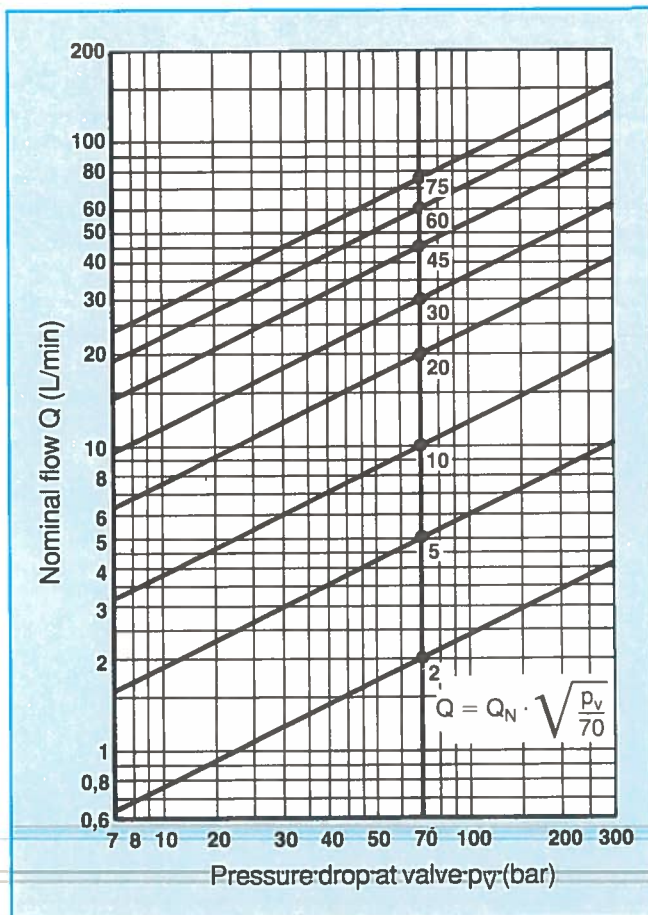


Fig. 16 Flow/load function of size 10 servo directional valves with barometric or electrical feedback (tolerance  $\pm 10\%$ )

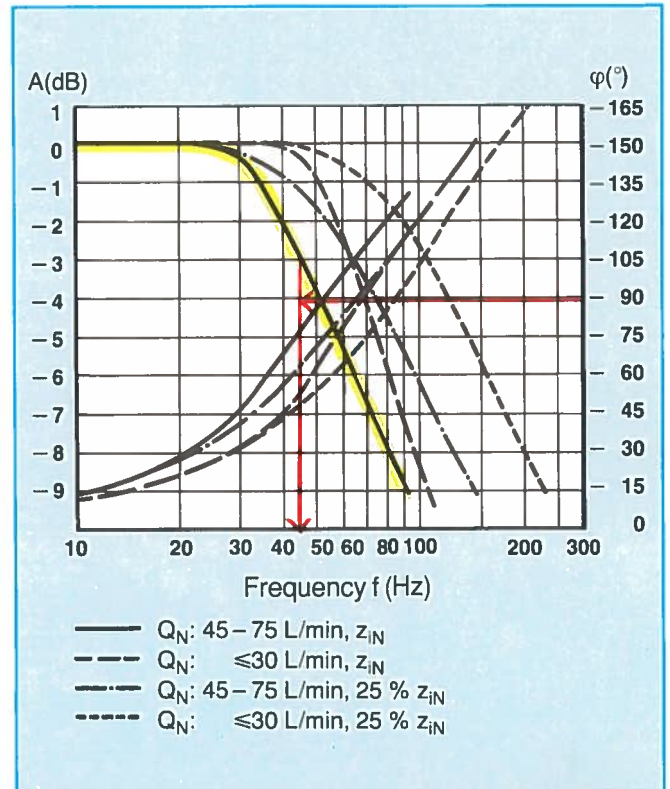


Fig. 17 Typical frequency response curve for directional servo valves with mechanical feedback

Comparison of the frequency response (Fig. 23) of size 10 servo directional valves with mechanical and barometric feedback shows that the servo directional valve with mechanical feedback has the better dynamic characteristics.

## 2-Stage with Barometric Feedback

The 2-stage servo directional valve basically consists of

- the 1st stage
- the 2nd stage with the interchangeable control sleeve (7), the control spool (3) and the control springs (4).

### 2nd Stage

The pressure difference between the two control chambers (8) and (9) of the control spool (3) is proportional to the electrical input signal of the 1st stage.

When power is not applied, the control spool (3) is pressure balanced and is held in the centre position by the control springs (4).

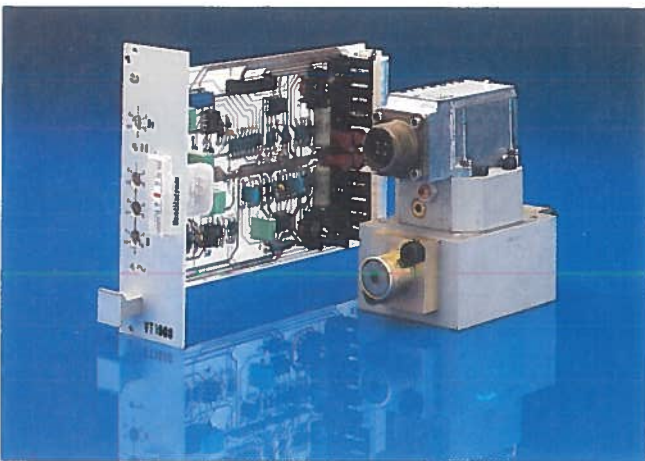


Fig. 18  
2-stage servo directional valve, Type 4 WS 2 EB 10

The flapper jet is offset by an electrical input signal, resulting in a pressure difference between the two control chambers (8) and (9).

The control spool is shifted until forces are balanced as the result of the pressure difference between the two control chambers (8) and (9) of the control spool (3) on the one side and the spring and flow force on the opposite side.

Since the control springs (4) also have a linear characteristic, the stroke of the control spool (3) and therefore the flow of the servo directional valve is also proportional to the electrical input signal.

### Special Valve Features

The connection dimensions of the main stage (2nd. stage) of this valve correspond to DIN 24 340.

The filter element in the 1st stage can be easily removed and serviced. The filter chamber prevents dirt particles entering the oil system.

An external pilot control is of advantage in particular valve applications. Since the DIN subplates feature no connection for this purpose, a subplate can be mounted between the 1st and 2nd stage.

The neutral point adjustment is easily accessible.

In servo valves with barometric feedback the amplitude drop and phase offset are dependent on the system pressure and flow. The devices are specially adapted for certain pressure ranges in order to achieve optimum results. The same also applies to certain flow ranges so that differing frequency curves are obtained.

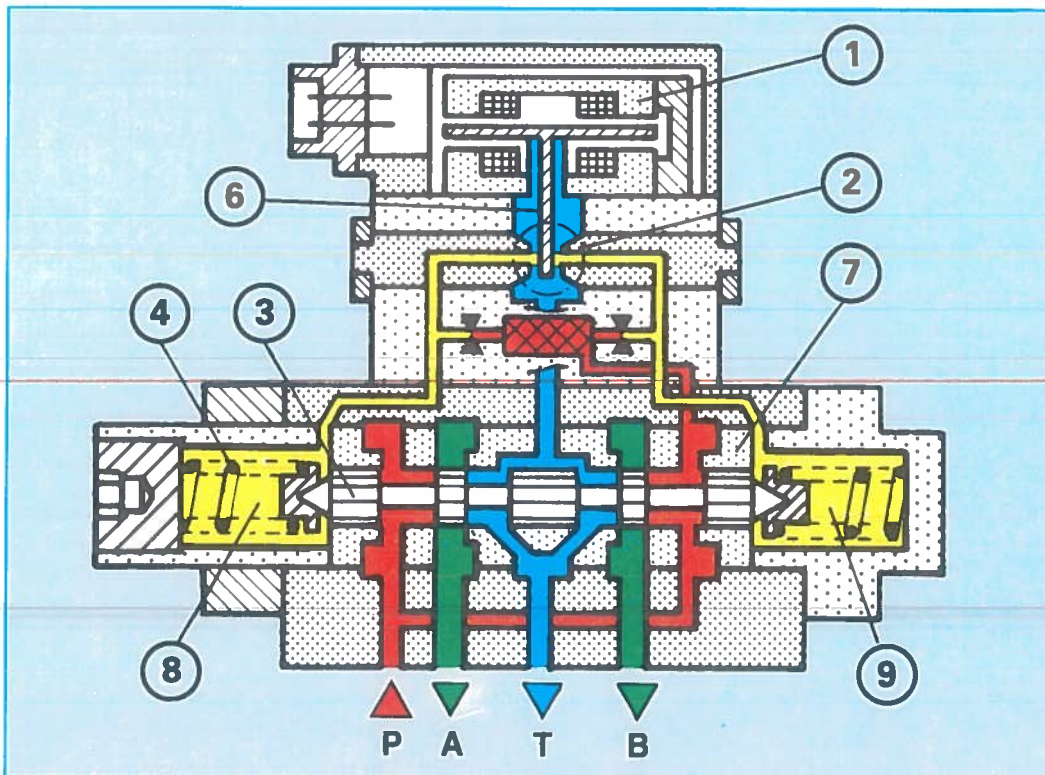


Fig. 19:  
2-stage servo directional valve with barometric feedback,  
Type 4 WS 2 EB 10

## 2-Stage with Electrical Feedback

Type 4 WS 2 EB 10 - 30/..B.. valves are 2-stage servo directional valves.

They basically consist of:

- the 1st stage
- the 2nd stage with the interchangeable control sleeve (3)
- an inductive positional transducer (4) with its core (5) secured to the control spool (6).

### 2nd Stage

The control spool (6) is linked to the inductive positional transducer (4) by means of suitable electronics.

Any change in the position of the main spool (6), which is attached to the core of the inductive positional transducer (5), produce a change in current in the AC fed coil (5). Such a change, or a change in signal value produces a resultant differential signal.

During signal/feedback comparison, the deviation is evaluated by suitable electronic components and fed to the first stage of the valve as a closed loop error. This signal deflects the flapper jet (7) between the two control jets (8). As a result, a pressure difference is produced between the two control chambers (9) and (10).

The control spool (6) with the attached core (5) of the inductive positional transducer (4) is shifted until the signal value agrees with the actual value and the flapper jet returns to the centre position.

In the controlled status, the control chambers (9) and

(10) are pressure-balanced and the control spool is held in this controlled position.

Due to the position of the control spool (6) with respect to the control sleeve (3) a corresponding control opening to flow results which is also proportional to the signal value in the same way as the spool stroke and the flow.

The frequency response of the valve is optimized by means of electrical amplification in the electronic circuitry.

### Special Valve Features

The connection dimensions of the main stage (2nd. stage) of this valve correspond to DIN 24 340.

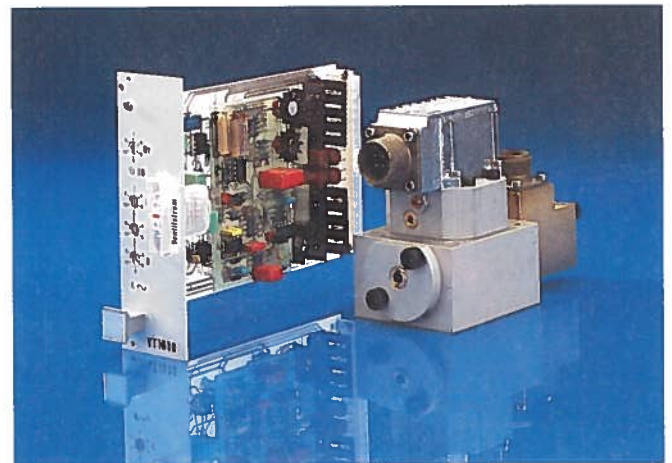


Fig. 20:  
2-stage servo directional valve, Type 4 WS 2 EE 10

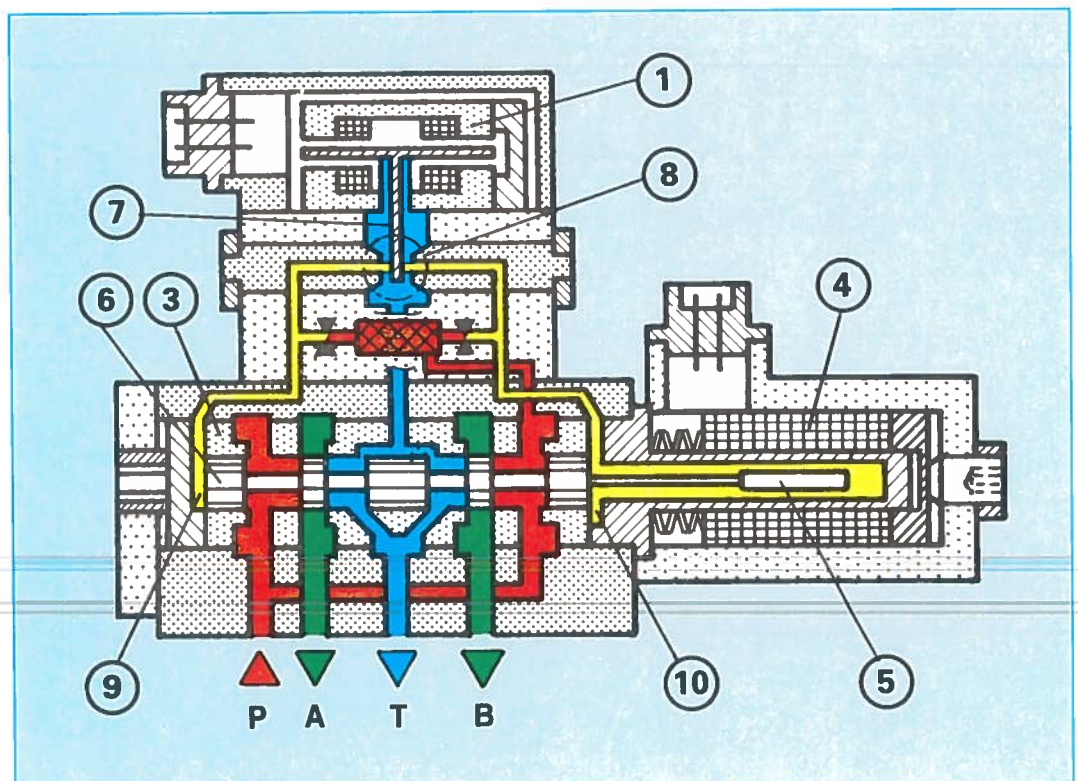


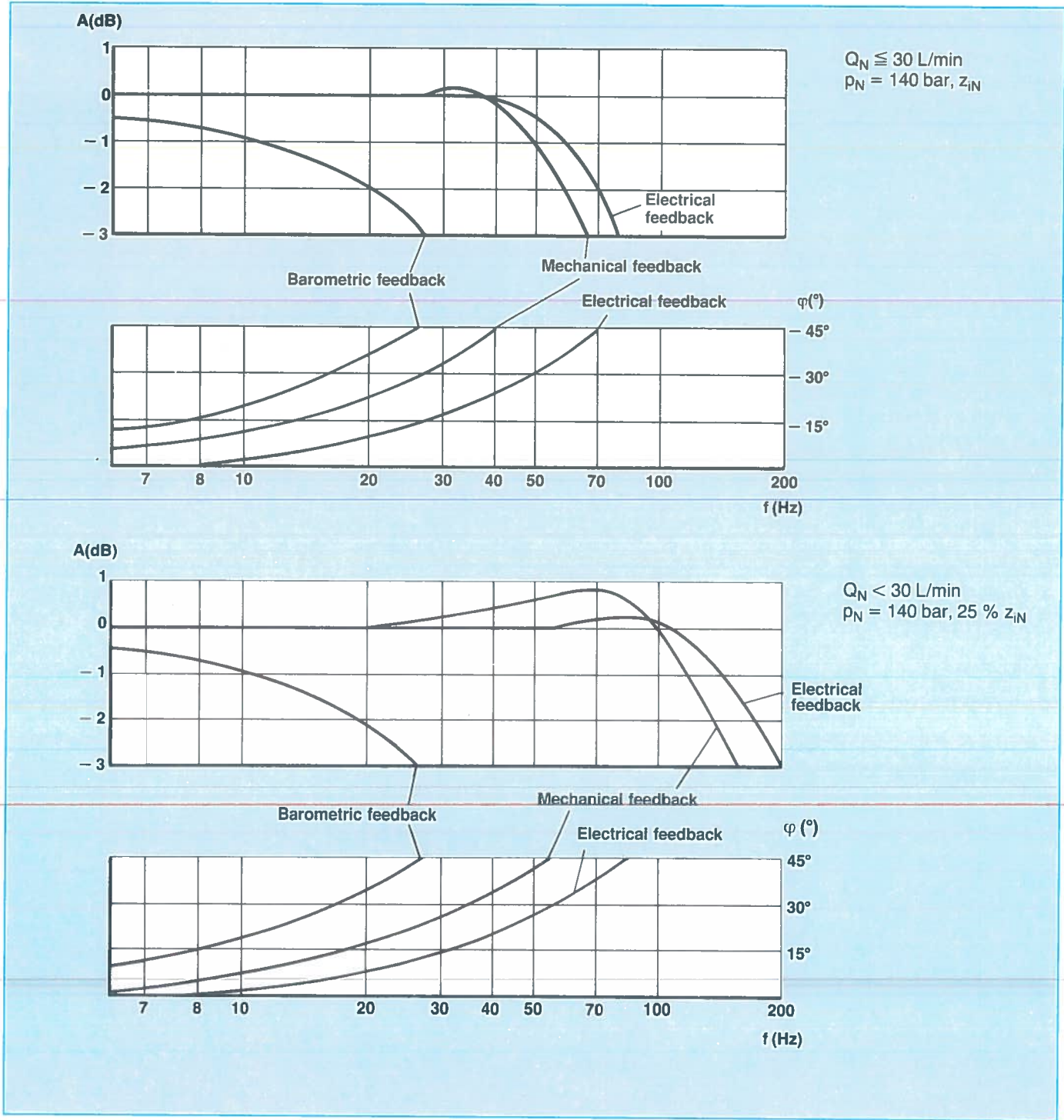
Fig. 21:  
2-stage servo directional valve with electrical feedback,  
Type 4 WS 2 EE 10

Comparison of the hydraulic and dynamic data shows the differences of the three feedback systems.

Feedback system	Mechanical Standard	Electrical	Baro- metric
Hysteresis dither-optimized (%)	≤ 2.0	≤ 0.5	≤ 3.0
Response sensitivity (%)	≤ 0.5	≤ 0.2	≤ 1.0
Reversal range (%)	≤ 1.0	≤ 0.2	≤ 2.0
Flow symmetry deviation (%)	≤ 5	≤ 5	≤ 5

Fig. 22 (right)  
Comparison of the hydraulic data

Fig. 23 (below)  
Comparison of the frequency response curves for  
mechanical, barometric and electrical feedback for size  
10 servo directional valves.



### 3-Stage with Electrical Feedback

Type 4 WS 3 EE .../... valves are 3-stage servo directional valves.

They basically consist of:

- the 1st stage
- the 2nd stage (3) in the form of a flow amplifier stage for control of the 3rd stage (4)
- the 3rd stage (4) for open loop flow control of the main oil flow
- an inductive positional transducer (5) with its core (6) secured to the control spool (7) of the 3rd stage.

#### 3rd Stage

The control spool (7) is linked to the inductive positional transducer (5) by means of suitable electronics.

Both the change in position of the control spool (7) as well as the change in the signal value generate via the core (6) a differential voltage in the coils of the positional transducer (5) fed with alternating current.

During signal/feedback comparison, the deviation is

evaluated by suitable electronic components and fed to the 1st stage of the valve as a closed loop error. This signal offsets the flapper jet (8) between the two control jets (9). As a result, a pressure difference is produced between the two control chambers (10) and (14). The control spool (11) is shifted and enables corresponding oil flow into the control chamber (15) or (16). The control spool (7) with the attached core (6) of the inductive positional transducer (5) is shifted until the signal value agrees with the feedback value.

In a controlled condition, the control chambers (15) and (16) are pressure balanced and the control spool is held in this controlled position.

Due to the position of the control spool (7) with respect to the control sleeve (13) a corresponding control opening to flow results which is also proportional to the signal value in the same way as the spool stroke and the flow.

The frequency response of the valve is optimized by electrical amplification in the electronic circuitry.

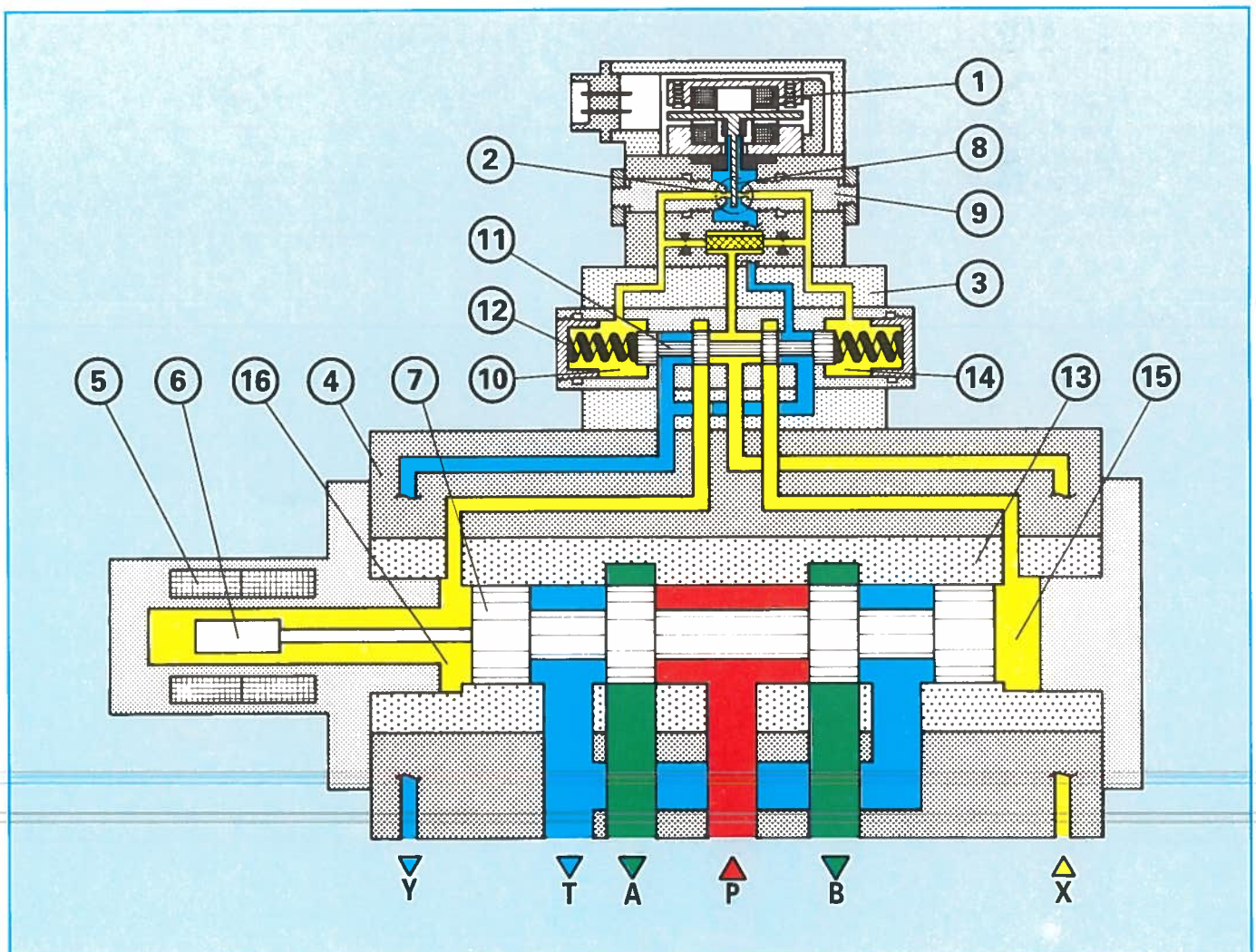


Fig. 24: 3-stage servo directional valve with electrical feedback, Type 4 WS 3 EE

## Servo control valve Size 6, direct operated

The control spool of this valve is not moved by a hydraulic pilot valve (jet/flapper) as in pilot operated servo valves, but mechanically by means of a powerful torque motor. This valve basically consists of the torque motor (1) and the 4-way spool stage (3).

The torque motor (1) is an electromechanical converter which converts an electrical signal into a linear motion of the end of the pin (4). It is hermetically sealed from the hydraulics. The armature (5), tube (6) and pin (4) are rigidly connected. The end of the pin (4) protruding from the motor is linked to the control spool (2) by means of the tie rod (7). The spring stiffness of the torque tube (6) acts against the force of the torque motor when the pin (4) is deflected so that a centering effect is obtained.

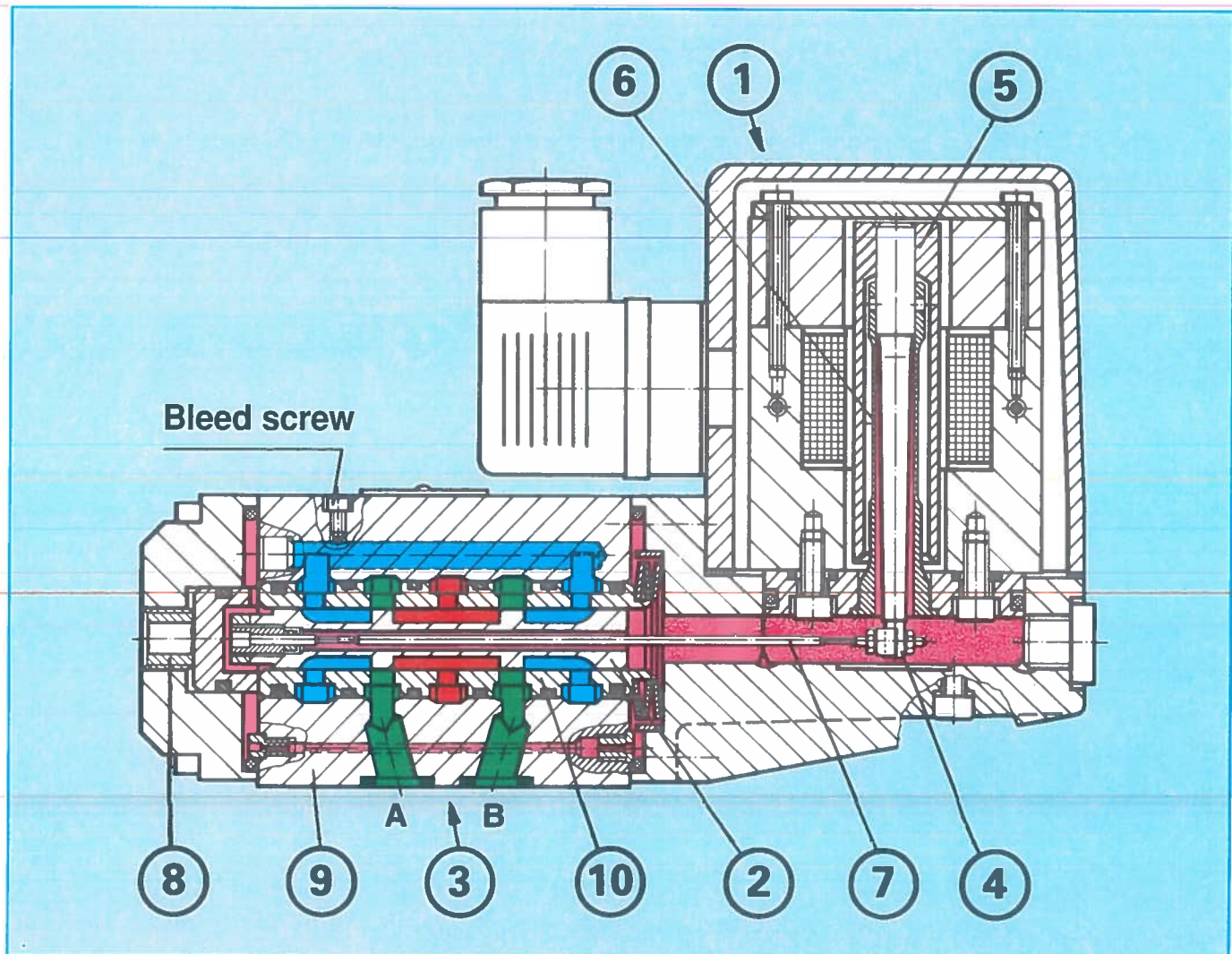
The movement of the control spool (2) and therefore the valve flow is proportional to the electrical input signal.

The hydraulic neutral point is adjusted with the screw (8) which moves the control sleeve (10), which can be shifted axially in the housing (9) relative to the control spool (2).

Special features of this "single-stage" control valve are:

- The permanent magnet (fast) motor which is both sealed and centered by the torque tube.
- Control sleeve and spool in "servo quality", i.e. linear flow curve, precise control land geometry.
- Hydraulic and electric damping.

Fig. 25 Single-stage control valve with permanent magnet torque motor actuation, Type 4 WS 1 EO 6



The stroke of the main spool is  $\pm 0.4$  mm; corresponding to the applied pressure drop, a flow/load function diagram is obtained as shown in Fig. 28. Since the motor acts against the flow forces with its positioning force only up to a certain limit, the main spool is gradually pulled back to the centre position at a certain " $\Delta p$ " despite the full input signal. As a result, the opening area is reduced and flow decreases.

On the other hand, this effect has a positive influence on the dynamics.

The smaller stroke is completed faster, a loss of amplitude due to the dynamic limits of the valve dependent on  $\Delta p$  occurs later than in the condition without flow.

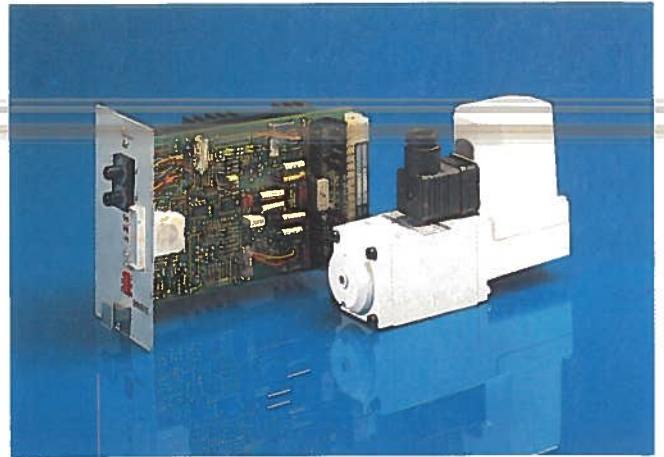


Fig. 26 Single-stage control valve, Type 4WS 1 EO 6

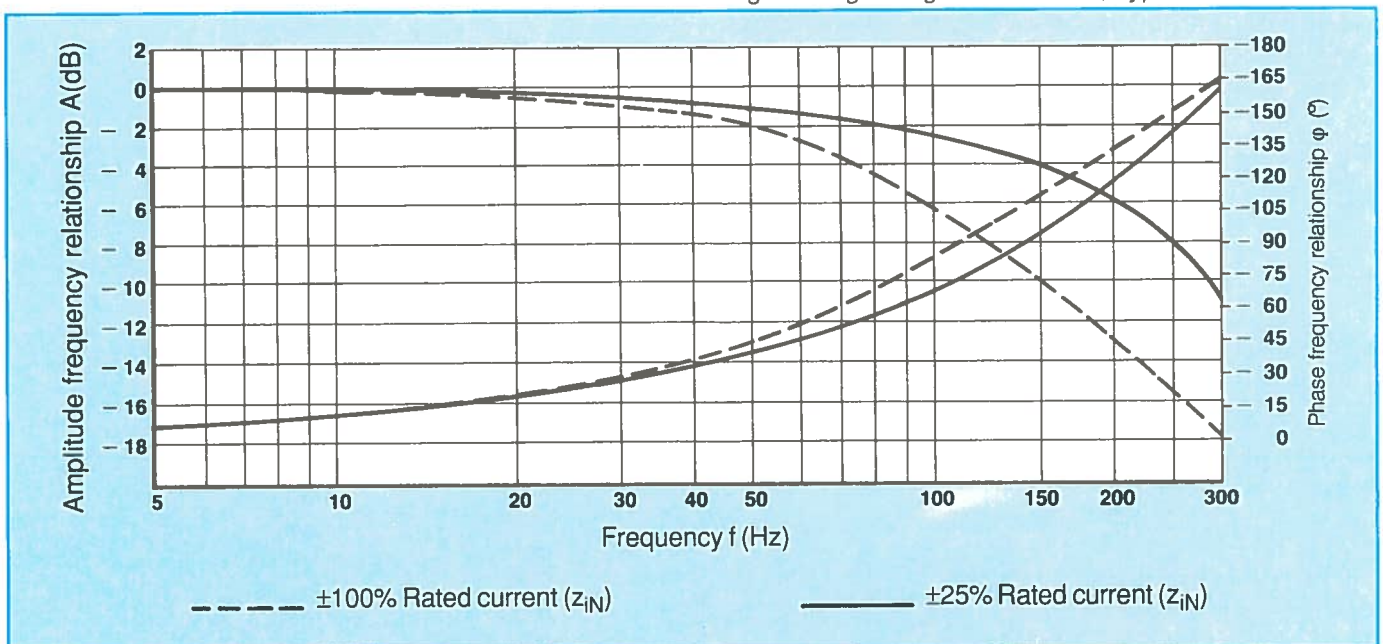


Fig. 27 Typical frequency response curve at  $p = 315$  bar and  $Q_N = 15$  l/min)

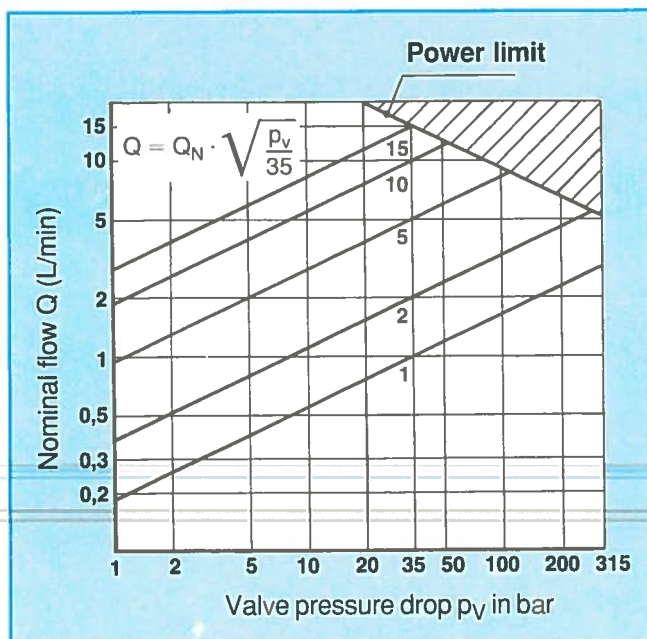


Fig. 28 Flow/load function at  $z_{IN}$  (the  $Q$ - $p_V$  relationship is negative above the power limit)

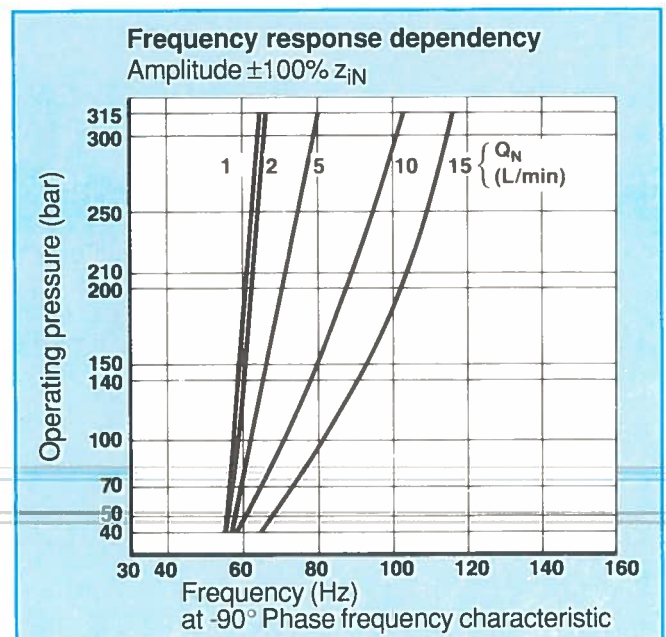


Fig. 29 Frequency response dependency (Amplitude  $\pm 100\%$   $z_{IN}$ )

## Proportional Directional Valve 4 WRV

The control valve described above with internal feedback can be used precisely in the same way as the 2-stage servo valve to control proportional valves with electrical feedback.

This 2-stage proportional valve is characterized by good dynamics and high repetition accuracy. In contrast to the 2-stage servo valve, the pilot valve does not require a continuous flow of pilot oil.

The valve is suitable for use in closed loop control circuits for force, speed and position control.

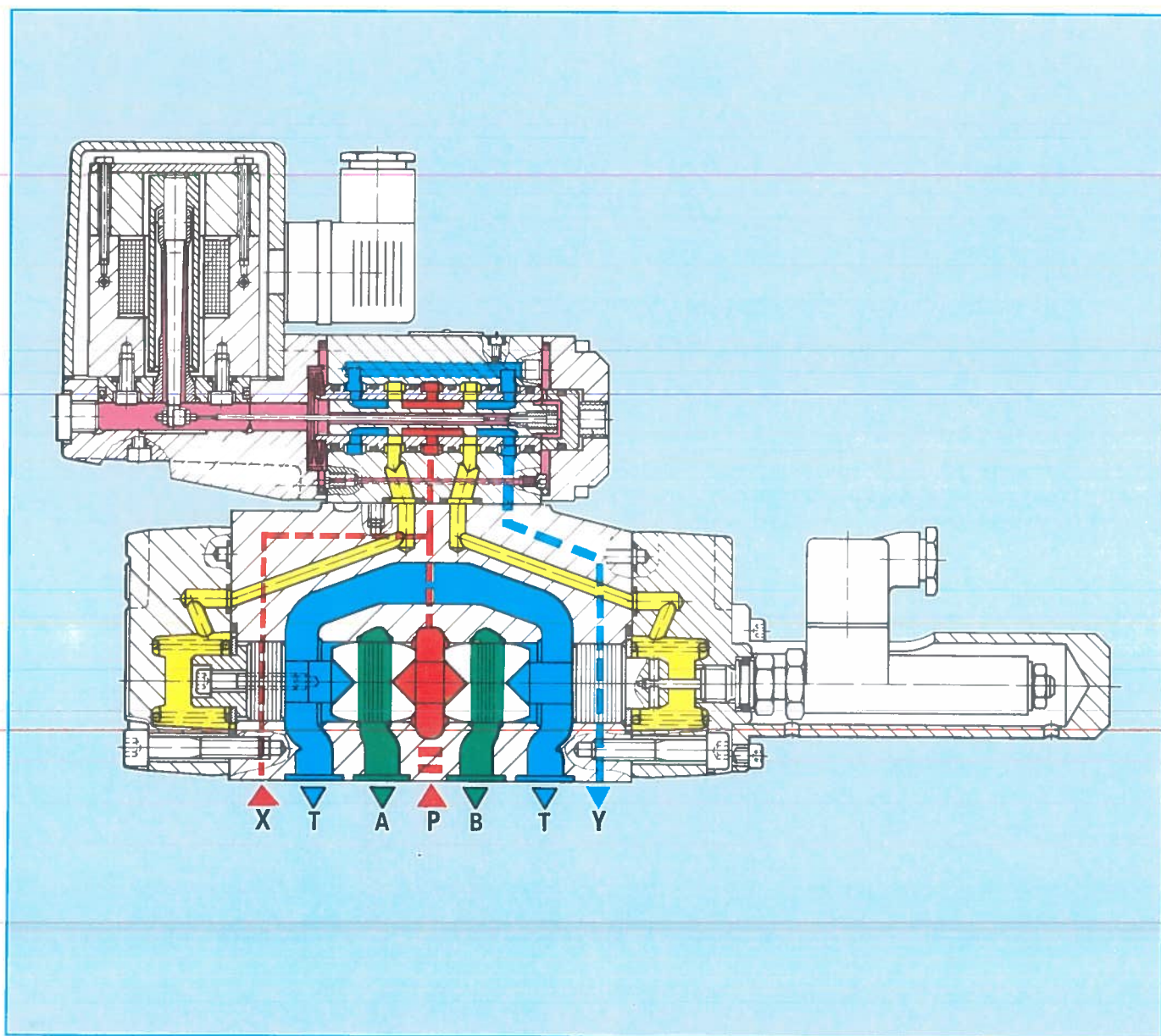


Fig. 30 Pilot operated proportional directional valve with electrical feedback, Type 4 WRV

## Installation, Commissioning and Maintenance of Hydraulic Servo Valves

### 1. General

The following should be observed to ensure perfect operation of the servo valves:

- The data sheets
- Cleaning and adjustment instructions provided in the servicing manual RD 09240
- We wish to further draw your attention to VDI Specifications, Commissioning and Maintenance of Hydraulic Systems, VDI (3027)

Note:

The functional test of each servo valve is verified by inspection data.

### 2. Installation

#### 2.1 Installation Instructions

Before the valve is mounted in the system, the valve type designation should be compared with the order data.

##### 1. Cleanliness

- of surrounding area and servo valve when mounting the unit.
- the tank must be sealed from external contamination.
- prior to installation, pipes and the tank must be cleaned of dirt, scale, sand, metal swarf etc.
- hot bent or welded pipes must be subsequently pickled, flushed and lubricated. Refer to the detailed information given at 3.6 with regard to system flushing.
- only use lint-free material or special paper for cleaning purposes.

2. Sealing material such as hemp, putty or sealing tape must not be used.

3. Hoses should be avoided whenever possible.

4. The piping system must be made up of seamless precision steel pipes to DIN 2391/C.

5. The connecting lines between the driven unit and the valve must be as short as possible; we advise that the servo valve is installed in the immediate vicinity of the driven unit.

The mounting surface must have a surface finish of  $Rt_{max} \leq 4 \mu m$  and a degree of flatness of  $\leq 0.01 \text{ mm}/100 \text{ mm}$  length.

6. The mounting screws must be tightened to the torque specified in the data sheet.

7. An oil bath air filter is recommended as the filter breather.

Pore size  $\leq 60 \mu m$ .

8. The protective plate on the servo valve should only be removed immediately before installation.

#### 2.2 Installation Position

Any position, preferably horizontal, however, the orientation of the spool must be observed related to the type of feedback. If the servo valve is mounted on the driven unit, care should be taken to ensure that the valve spool is not positioned parallel to the acceleration direction of the driven unit.

#### 2.3 Electrical Connection

Refer to the relevant data sheet for electrical connection.

The servo valve can be operated in parallel, series or differential circuits. To provide the highest degree of operational reliability, we recommend parallel connection.

Caution:

Due to the electrical amplification in closed loop control circuits, an electrical signal must not be fed to the valve before operating pressure is applied to the first stage. An exception to this is when 100 % current limitation is provided.

Special protection classes require special measures which are stipulated in the relevant data sheet.

### 3. Commissioning

#### 3.1 Fluids

Mineral oil to DIN 51524, DIN 51525 or VDMA 24318 should preferably be used as the pressure medium. A fluid temperature of 50 °C should be maintained when using H-L36 or H-LP36. The maximum temperatures recommended by the manufacturer of the fluid should, if possible, not be exceeded in order to protect the fluid. To ensure constant control characteristics of the system, it is advisable to maintain the oil temperature constant ( $\pm 5 \text{ }^{\circ}\text{C}$ ).

Other fluids on request.

#### 3.2 Are the correct sealing materials being used?

For "non flam" fluids type HFD, and for temperatures above 90 °C, seals type "V" must be specified.

#### 3.3 Filtration

- Internally pilot operated servo valves must be protected immediately in front of the valve with a pressure line filter without bypass valve with a nominal filter element pore size of 10  $\mu m$  (purity class 5 to NAS 1638) in the pressure port "P".

- In the case of externally pilot operated valves a pressure line filter without bypass and nominal pore size of the filter element 10 µm (purity class 5) must be installed before the servo valve in the supply line to port "X". In this case, we recommend that the entire hydraulic system is cleaned via a further 10 µm filter.

- The permissible differential pressure of these filters must be greater than the operating pressure.

- We recommend filters equipped with a filter clogging indicator.

- Absolute cleanliness must be ensured during filter change. Impurities at the output side of the filter are flushed into the system and cause faults and malfunctions.

Impurities at the inlet side reduce the service life of the filter element.

**3.4** Due to requirements regarding good control characteristics the pilot pressure should be maintained constant (±5 bar).

**3.5** Adjustment of the hydraulic neutral point:

The hydraulic neutral point of each servo valve is adjusted on a test bench with the aid of a hydraulic motor. Nevertheless, in order to achieve optimum closedloop control accuracy it may be necessary to readjust the hydraulic neutral points of the servo directional valves corresponding to the relevant load and in accordance with the instructions given in the data sheet.

**3.6** Flushing system:

All supply and return lines must be flushed prior to commissioning of the servo valve. Instead of flushing plates which connect P to T (refer to data sheet for type), the use of directional valves (symbol G or H) is preferred, with the aid of which the working lines and loads can also be flushed. Care must be taken in the case of external pilot oil connection to ensure that this connection is also flushed.

The oil in the system should be flushed through the filter at least 150 ... 300 times.

Based on this, a reference flushing time can be derived:

$$t = V/Q \cdot 2.5...5$$

Were  $t$  = Flushing time in hours  
 $V$  = Tank volume in liter  
 $Q$  = Pump delivery rate in l/min

During the flushing procedure, all filters must be constantly monitored and the filter elements replaced if necessary. After opening connection lines (for any reason whatsoever) flushing must be continued for approx. 30 minutes.

## 4. Maintenance

**4.1** The system must be flushed again when refilling more than 10 % of the tank capacity (see 3.6).

**4.2** Return of Valve for Servicing

When returning a defective valve, it is necessary to protect the base surface of the valve from the effects of dirt. Careful and adequate packing is advisable to ensure no further damage is incurred during transport.

**4.3** Cleaning and Adjustment Instructions

Past experience has shown that faults in servo valves are mainly due to dirt in the area of the flapper jet system. Valves can be cleaned in accordance with the instructions given in the Service Manual RE 09420.

## 5. Storage

Servo valves must be stored in a dry, dust-free room with low atmospheric humidity. Storage rooms must also be free of corrosive materials and vapours. The valves must be checked from time to time to ensure that they are stored correctly. If servo valves are to be stored for longer than 3 months, it is advisable to fill the valves with a preservative oil.

**Notes**

Notes

Chapter H

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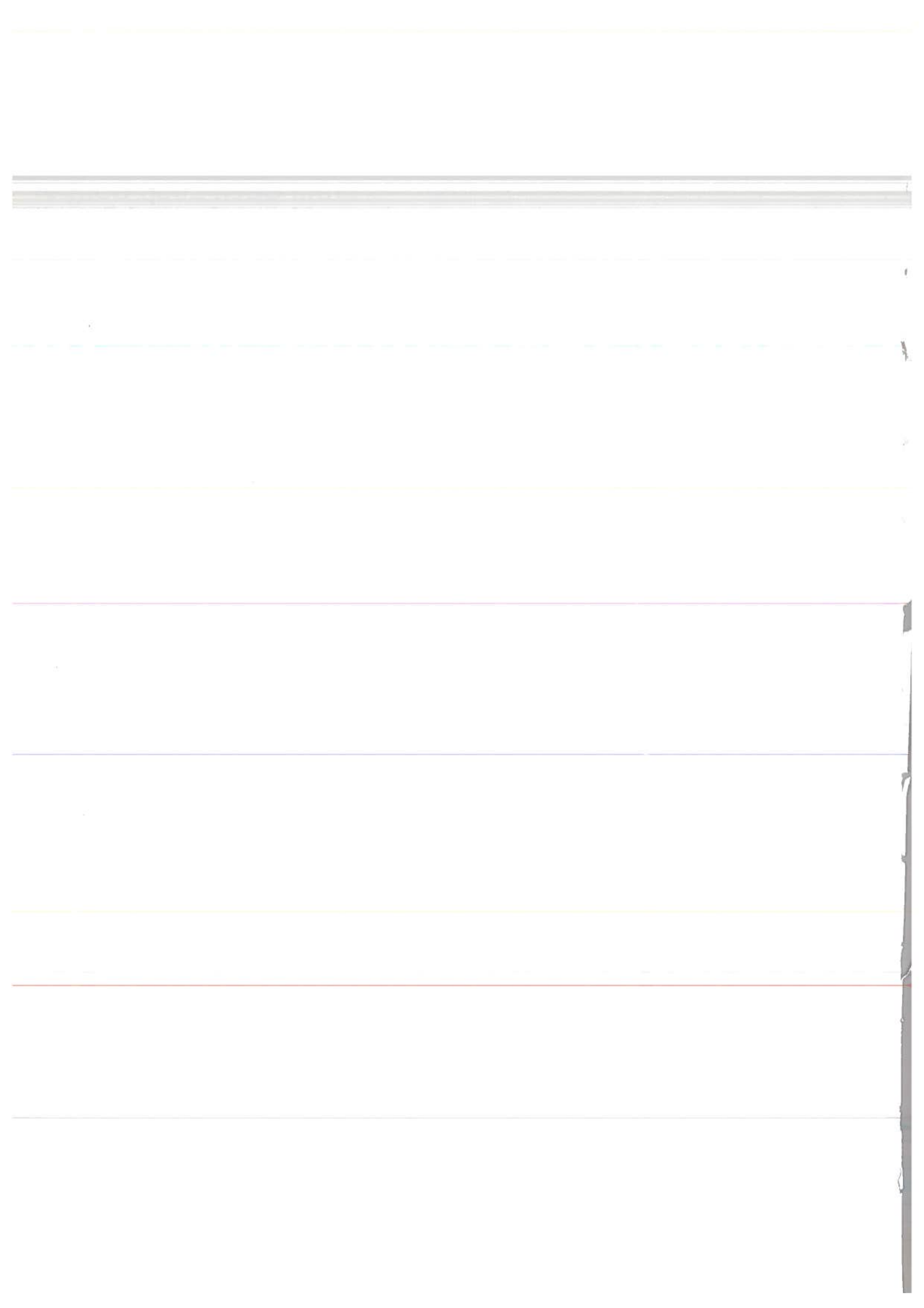
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# **From Open Loop Control to the Closed Loop Control Circuit**

Arno Schmitt, Dieter Kretz

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## From Open Loop Control to the Closed Loop Control Circuit

As shown by calculation examples for the design of open loop control systems with proportional valves, the possible accuracy of the system depends on several factors resulting from the overall system.

Before dealing with closed loop control circuits in detail, we should take a look at two types of control systems as an introduction:

- time-dependent deceleration
- distance-dependent deceleration

### 1 Time-dependent Deceleration

The following relationship results if an electrical time ramp is used in a open loop control with proportional valves for the deceleration procedure:

#### 1.1

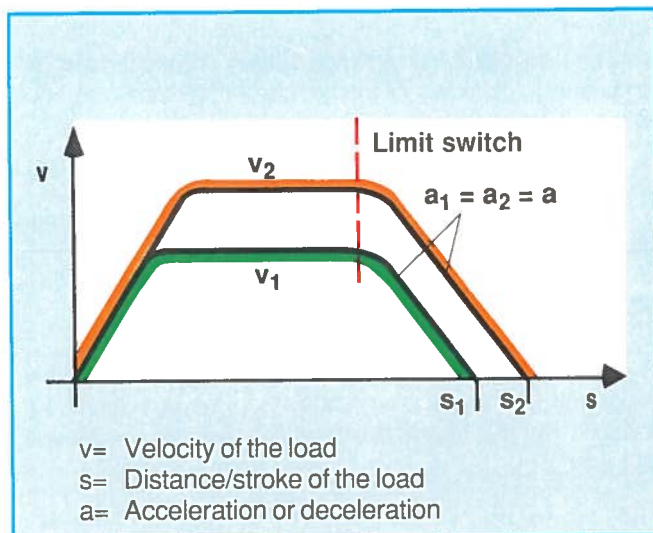


Fig. 1

A cylinder moves at velocity  $v_1$ . On reaching the limit switch the selected velocity value (deflection of the valve spool) is switched at the input of the card to  $v = 0$  for instance, i.e. cylinder stop. The signal value now changes corresponding to the ramp time setting. The deceleration distance is thus determined.

Example:

$$\begin{aligned} v_1 &= 0.8 \text{ m/s travel speed} \\ t_{b1} &= 0.2 \text{ sec. deceleration time} \\ \rightarrow a &= v/t \\ a &= 0.8 \text{ (m/s)}/0.2 \text{ (s)} = 4 \text{ [m/s}^2\text{] deceleration} \end{aligned}$$

#### Deceleration distance

$$s_1 = v_1^2 / (2 \cdot a) = 0.8^2 / (2 \cdot 4) = 0.08 \text{ [m]} = 80 \text{ [mm]}$$

If, for example, the velocity is changed corresponding to the required operation, a different deceleration distance naturally results with the ramp setting remaining unchanged.

Example:

$$\begin{aligned} v_2 &= 1.2 \text{ m/s travel speed} \\ t_{b2} &= 0.3 \text{ sec. deceleration time} \\ \rightarrow a &= v/t \\ a &= 1.2 \text{ (m/s)}/0.3 \text{ (s)} = 4 \text{ (m/s}^2\text{) deceleration} \end{aligned}$$

#### Deceleration distance

$$s_2 = v_2^2 / (2 \cdot a) = 1.2^2 / (2 \cdot 4) = 0.18 \text{ [m]} = 180 \text{ [mm]}$$

This therefore means the cylinder stops at various points. This fact is often forgotten in practical applications when a stopping point is approached at various velocities.

#### 1.2

A possibility of reaching a stopping point at various velocities is to decelerate to a relatively low speed. The limit switch E2 only send the stop signal outside this velocity range. Fig. 2 shows the progression. The stopping accuracy is good in this case (also see Page E16).

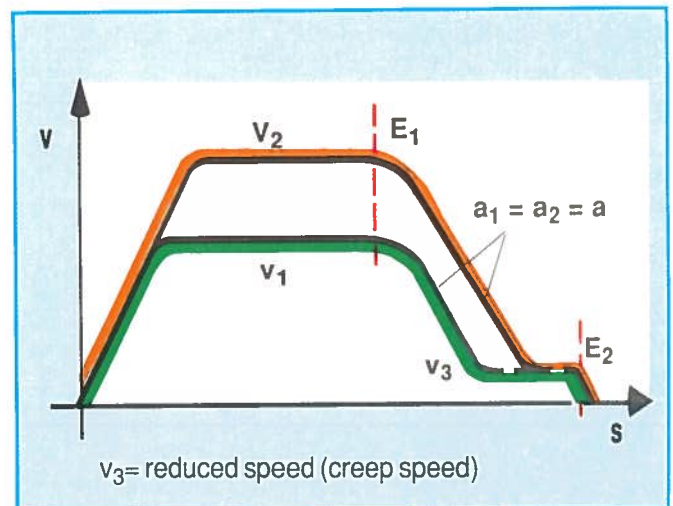


Fig. 2

However, at these velocities  $v < v_{\max}$  this arrangement takes valuable time.

**1.3** A further possibility is to assign a ramp to each velocity value. If, for example, the same stopping point is to be approached from various speeds, theoretically this can be achieved as follows:

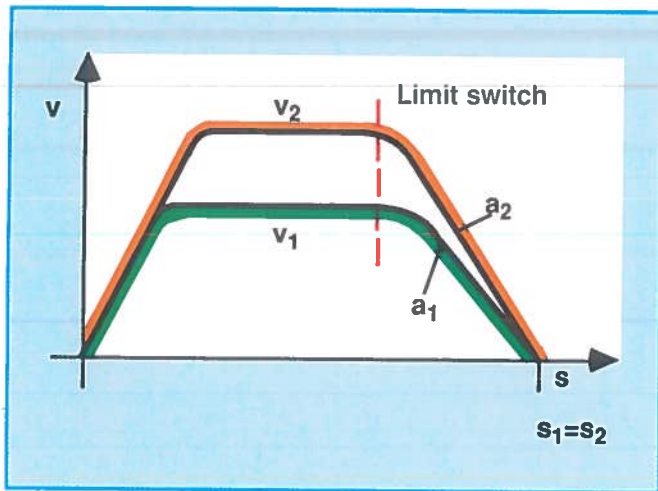


Fig. 3

Although the same deceleration distance is obtained at the corresponding ramp setting, time is however lost (also as in Example 1.2, see Page E16).

**Example:**

Let us take the previously calculated deceleration distance of 180 mm (at  $v_2 = 1.2 \text{ m/s}$  and  $a = 4 \text{ m/s}^2$ ) this results in

$v_1 = 0.8 \text{ m/s}$  and  $s_b = 180 \text{ mm}$

→ a deceleration of

$$a_1 = v^2 \cdot 10^3 / 2 \cdot s = 0.8^2 \cdot 10^3 / 2 \cdot 180 = 1.8 \text{ [m/s}^2\text{]}$$

and the required time of

$$\rightarrow t_b = v/a = 0.8/1.8 = 0.44 \text{ [sec]}$$

In practical applications, the scatter at the end point is greater than in Example 1.2 since movement is always from different speeds.

In this connection, reference is once again made to the maximum possible acceleration/deceleration as specified on Page E27/E28.

The problems with regard to the accuracy of the ramp setting should also be taking into consideration at this point so that this solution is not really recommendable for applications where particular importance is attached to an exact stopping point.

**1.4** In order to achieve a more reliable deceleration of the system compared to Example 1.3, it would be necessary to use a further limit switch for the other velocity:

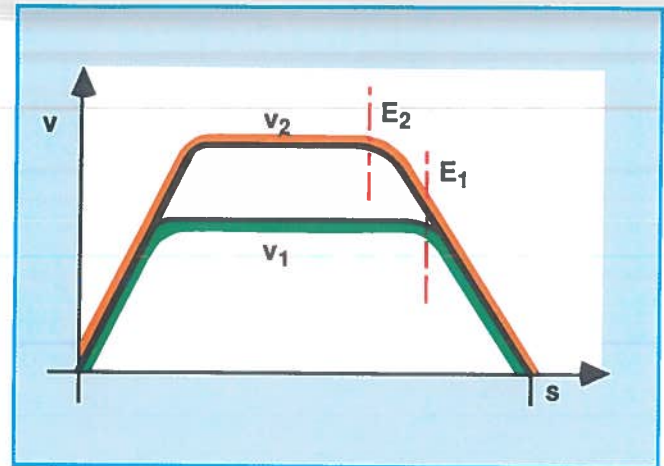


Fig. 4

The limit switch E1 is arranged later corresponding to the lower velocity  $v_1$ . In this solution therefore one limit switch is assigned to each velocity value.

The solution which enables the characteristics shown in Fig. 4 without having to use a separate limit switch for each velocity is distance-dependent deceleration.

## 2 Distance-dependent Deceleration

Clearly by way of definition, deceleration takes place not dependent on an electrical ramp time but rather dependent on the travel distance (stroke) of the driven unit.

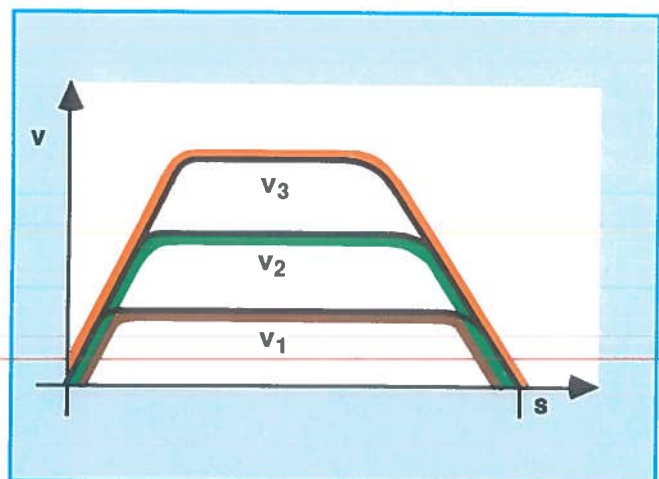


Fig. 5

The diagram in Fig. 5 clearly shows that the same stopping point is always reached irrespective of the travel speed.

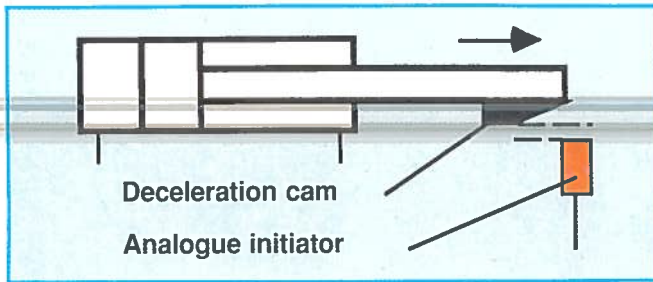


Fig. 6

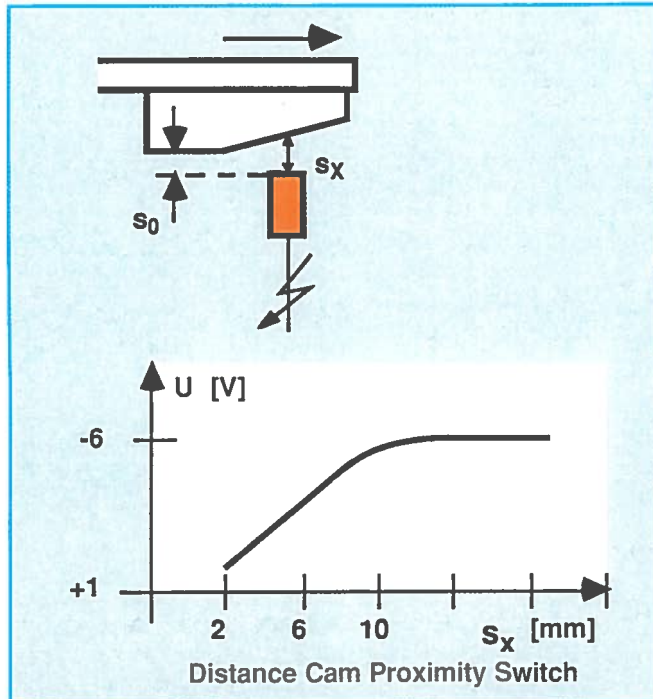


Fig. 7

A version often used in practical applications for stroke-dependent deceleration involves a deceleration cam and analogue proximity switch (Fig. 6).

The analogue initiator is an electronic proximity device, which produces a voltage output inversely proportional to the proximity of a metal component (i.e. a cam), from maximum down to virtually 0 volts. This voltage is routed to a specially designed amplifier and controls the proportional solenoids of the proportional valves.

The block diagram (Fig. 9) shows the control with analogue initiator. For the sake of simplicity, only a solenoid control system is shown.

The minimum value comparator ensures only the smaller of the two input signals ( $E_1$  = signal value,  $E_2$  = from proximity switch) is effective at the output.

As also shown in the block diagram, a root value generator (Fig. 8) is often used in conjunction with the analogue initiator. The advantage for practical applications is that time requirements are reduced since this arrangement enables optimum approach of a position, i.e. at the highest possible speed. A description of an implemented system is provided on Pages L8 and L9.

If the analogue positional feedback is to be effective only in the area of the deceleration distance (always the same end point), systems can be designed without the necessity of taking the whole travel distance into consideration.

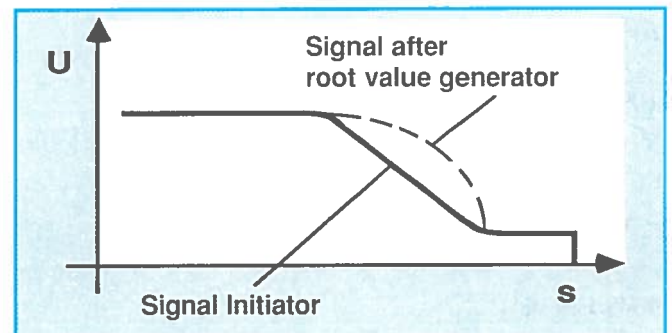


Fig. 8

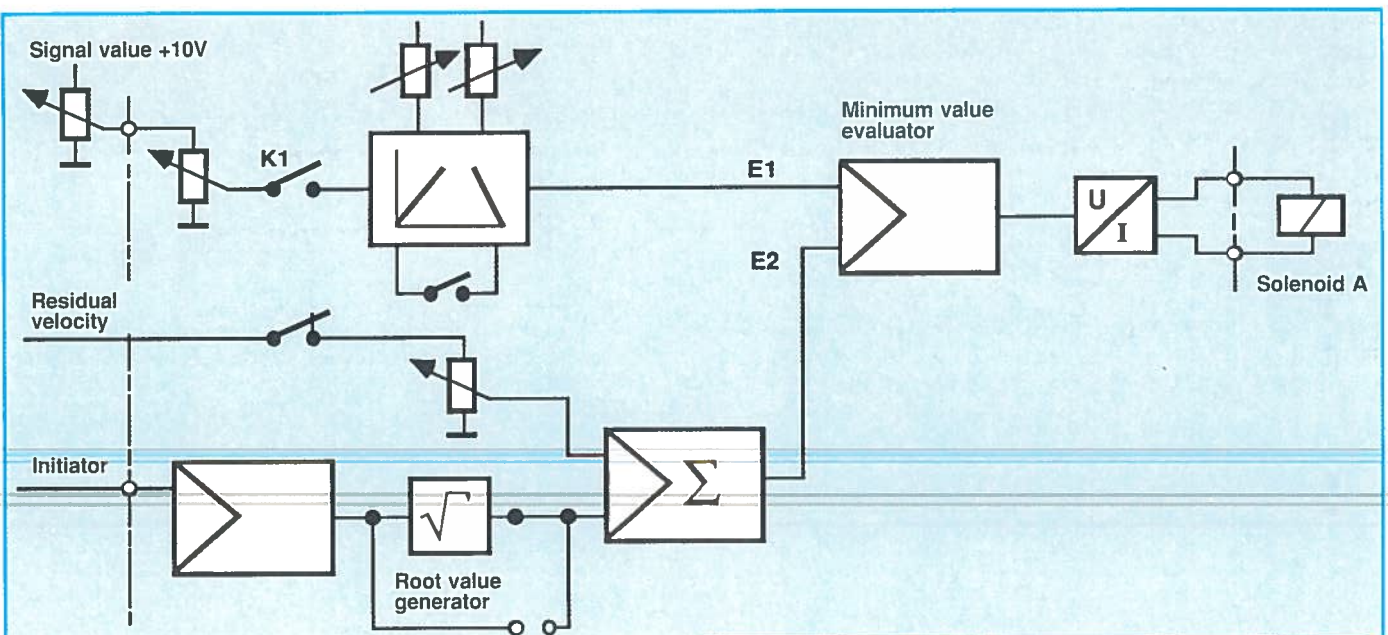


Fig. 9 Practical control with analogue initiator

Another possibility of determining position during stroke-dependent deceleration is a linear potentiometer.

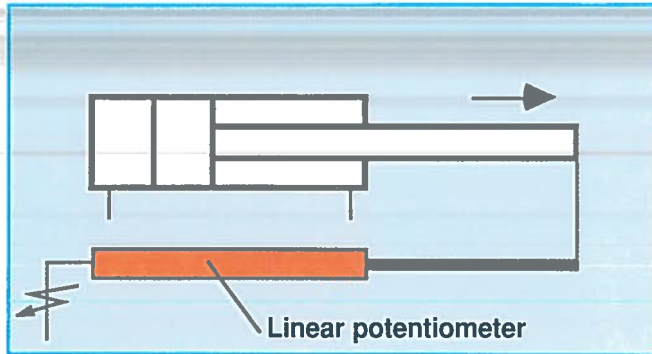


Fig. 10

In this version, an analogue signal of stroke position is taken and processed on an electronic amplifier card.

Since, in this case, the entire stroke is converted in the form of a signal, it is possible to preselect any stroke with the electrical amplifier.

The examples described up till now are clearly open loop controls.

This means that the actual value, e.g. velocity of a cylinder, is not measured and not compared with the signal value.

In such systems, the interfering variables naturally have an effect on the result.

If it is necessary to compensate these disturbing influences, the system must be designed as a closed loop control circuit.

## The Closed Loop Control Circuit

The prerequisite to facilitate understanding of the relationships in a closed loop control circuit is the knowledge of fundamental conditions and definitions relating to closed loop technology.

A general view of the most important relationships is provided in the following.

Here, the intention is not to provide information relating to formulae and methods of calculation, but rather to convey the fundamentals of physical relationships in the language of the control technician.

### What is Closed Loop Control?

Fig. 11 shows the principle layout of a closed loop control circuit together with the most important definitions.

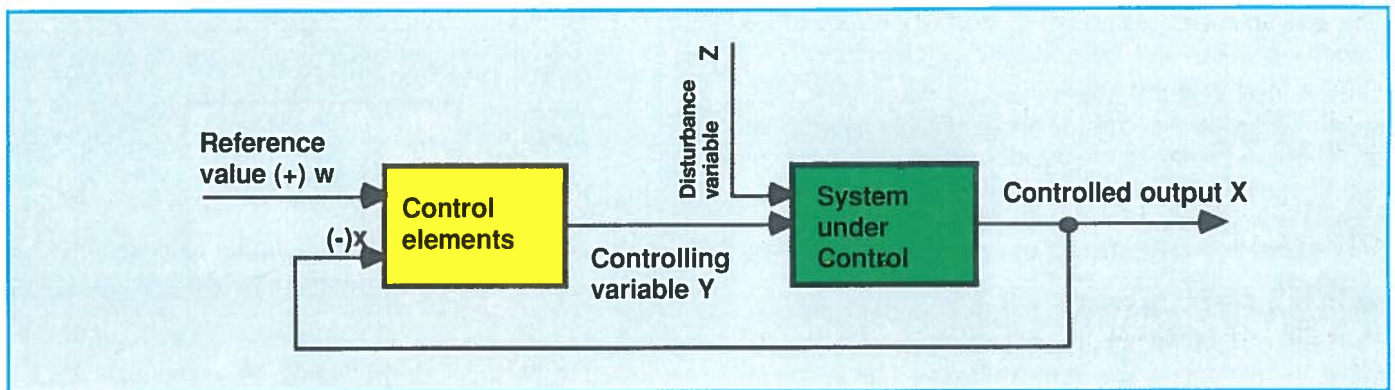


Fig. 11 Principle configuration of a closed loop control circuit

### Definition

In a closed loop control system, the feedback value is constantly measured and compared with a signal value. As soon as a difference occurs between the two values caused by an interfering variable, a corresponding adjustment is implemented in the system to be controlled with the aim of re-establishing agreement of the control variable with the signal value.

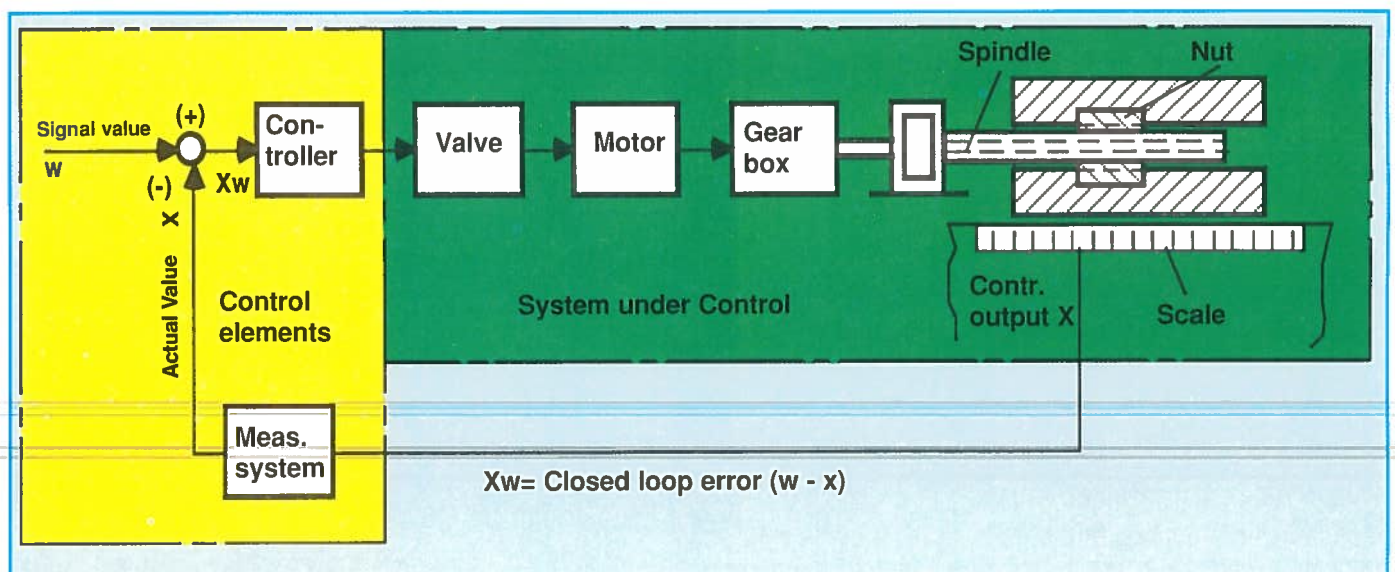


Fig. 12 Example of closed loop positional control

As in any other closed loop control system, the closed loop positional control circuit features a control unit and a controlled system.

In the example (Fig. 12) the control equipment includes:

- the controller

This consists of the comparator which generates the signal value/feedback difference, and the closed loop amplifier

- the precision measuring system.

The controlled system includes:

- the hydraulic drive with hydraulic motor and valve
- the mechanical transmission elements

such as

- gearbox
- coupling
- threaded spindle

The characteristic feature of a closed loop control is looped signal flow, which works here as follows:

The position  $X$  of the slide (= control variable) is measured with the aid of a scale and a measuring amplifier and represents the actual position value. The set position is specified by the signal value  $w$  produced by a signal value generator. The closed loop error is derived from the difference between the signal and actual value ( $w - x$ ).

The closed loop error is passed through the controller. The output signal of the controller is the positioning variable  $Y$ . This positioning variable  $Y$  also acts as the input variable of the controlled system and actuates the valve. The direction of rotation of the motor is converted by means of a spindle drive into a linear motion of the slide. In this way, the signal flow is closed to form a closed loop positional control circuit.

### Block Diagram

The individual areas of the closed loop control circuit such as "controlled system" and "control equipment" are termed "closed loop control elements". These closed loop control elements are generally represented in the form of rectangular blocks.

The block diagram reflects the link between the individual blocks to form a closed effective system.

The signal flow is represented by lines and direction arrows.

### Transfer functions

Input signals or "input variables"  $X_w$  act on the individual elements in the closed loop control circuit. The "output variables"  $X_a$  are derived from these values depending on the "transfer function" of the element and then further processed.

The "transfer function" represents the progression of the output variable with time referred to any time change in the input variable.

A characteristic change in the input variable is the step function. In this case, the output signal is the "stepped response" or the "transfer function".

This transfer function is often drawn in the block symbol to facilitate exact or more illustrative representation of the transfer function of an element.

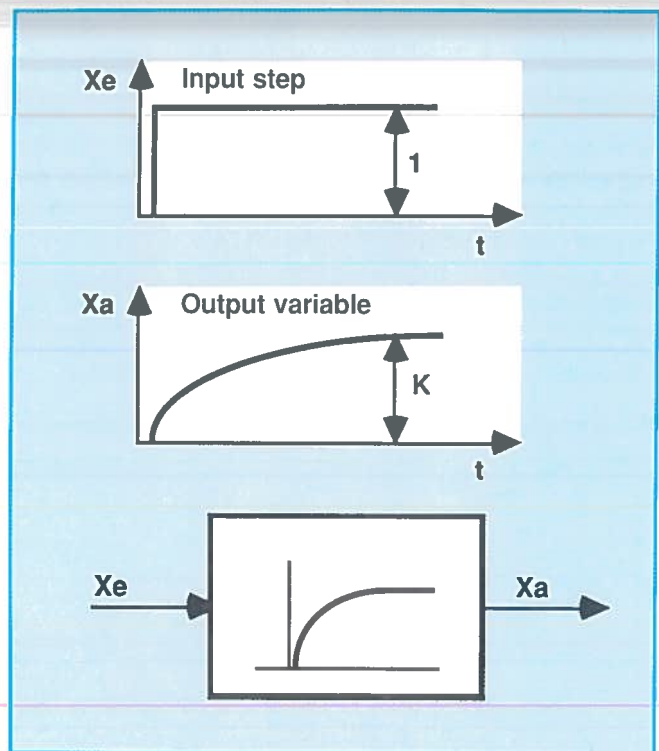


Fig. 13 Example of a transfer function

Despite the wide variety of components possible from a technical point of view, the various transfer function can be reduced to only a few basic types.

This exclusion of component diversity in the transition from the real technical system to the mathematical model greatly facilitates examination of dynamic procedures and generally enables statements to be made relating to the characteristic of a closed loop control, irrespective of whether the closed loop control circuit comprises electrical, mechanical or any other components.

The closed loop control elements can be categorized in "basic transfer functions" (Fig. 14) according to their transfer characteristic.



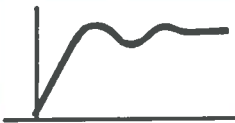



1.) Proportional element		P element
2.) Proportional element with delay 1.order		PT <sub>1</sub> element
3.) Proportional element with delay 2.order		PT <sub>2</sub> element
4.) Integral element		I element
5.) Differential element		D element
6.) Dead time element		T <sub>dead</sub> element

Fig. 14 Basic transfer functions

### Examples relating to basic transfer functions

#### The proportional element (P-element)

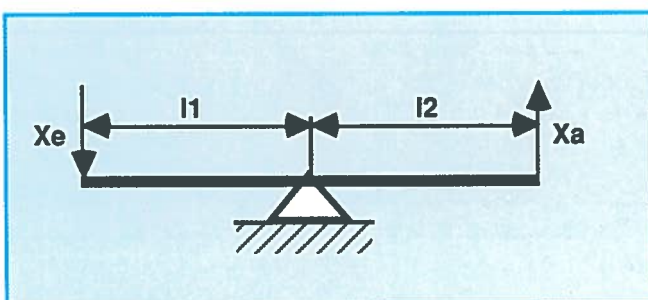


Fig. 15

The stepped change in the output variable  $X_a$  corresponds to the stepped change in the input variable  $X_e$ .

The output value is

$$X_a = X_e \cdot l_2 / l_1 = K \cdot X_e$$

with the amplification of the proportional element (also termed transfer constant).

$$K = l_2 / l_1$$

This provides the symbol for the P-element

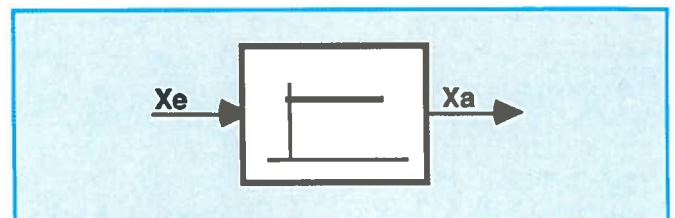


Fig. 16 Symbol for P-element

Further examples of the effect of the P-element are the relationship  $U = R \cdot I$  between current  $I$  and voltage  $U$  referred to an ohmic resistance  $R$ , or the relationship  $F = m \cdot a$  between acceleration  $a$  and force  $F$  at an accelerated mass  $m$ , or the ideal amplifier with resistance circuit (refer to appendix for explanation).

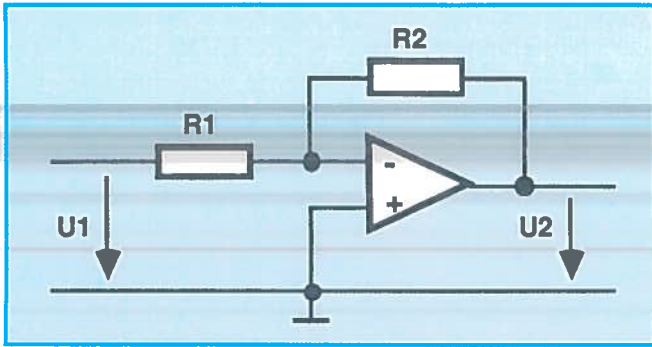


Fig. 17 Amplifier with resistance circuit

The stepped change in the output voltage  $U_2$  corresponds to a stepped change in the input voltage  $U_1$ .

The output voltage is

$$U_2 = -R_2/R_1 \cdot U_1 = K \cdot U_1$$

with amplification

$$K = R_2/R_1$$

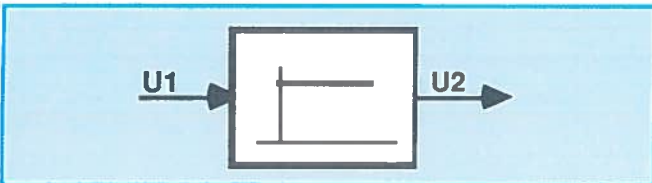


Fig. 18

## Integral Element (I-element)

Linear increase of the output signal with respect to time.

$$X_a = K \cdot \int X_e(t) \cdot dt$$

Also in this case,  $K$  is termed the transfer constant or the amplification factor of the I-element.

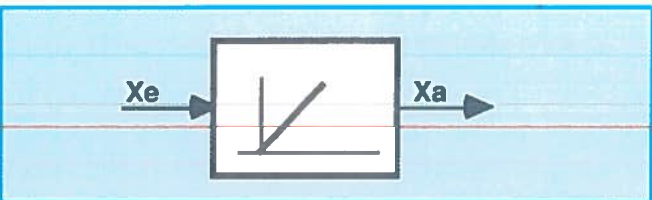


Fig. 19 Symbol of the I-element

Example of the effects of the I-element: Hydraulic cylinder

Stroke  $s$  moved depends on the oil flow  $Q$

$$s = 1/A \cdot \int q \cdot dt$$

with  $K = 1/A$   $A$  = effective area

or: Hydraulic motor

The angle of rotation of a motor shaft depends on the angular velocity  $\omega$ .

$$\varphi = K_0 \cdot \int \omega \cdot dt \quad K=1$$

or: Spindle drive

Conversion of a spindle speed  $n$  into linear motion.

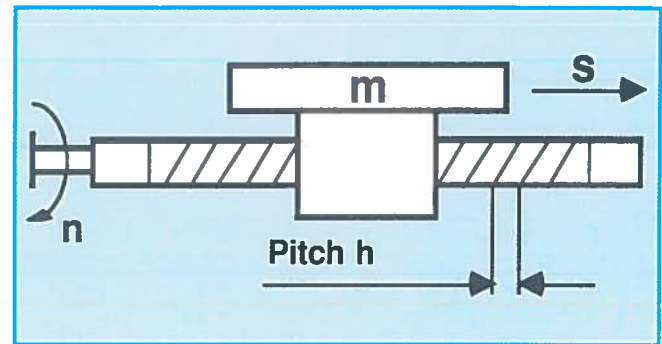


Fig. 20

The stroke  $s$  as the output value is:

$$s = h \cdot \int n \cdot dt$$

at constant spindle speed  $n$  the stroke  $s$  is

$$s = h \cdot n \cdot t$$

i.e. linear increase in stroke with respect to time.

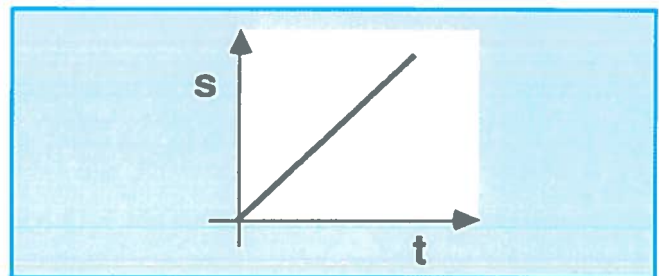


Fig. 21

## Differential Element (D-element)

$$\begin{aligned} X_a(t) &= K \cdot \dot{X}_e(t) \\ \dot{X}_e(t) &= dX_e/dt \end{aligned}$$

The value of the output signal depends on the change in velocity of the input signal.

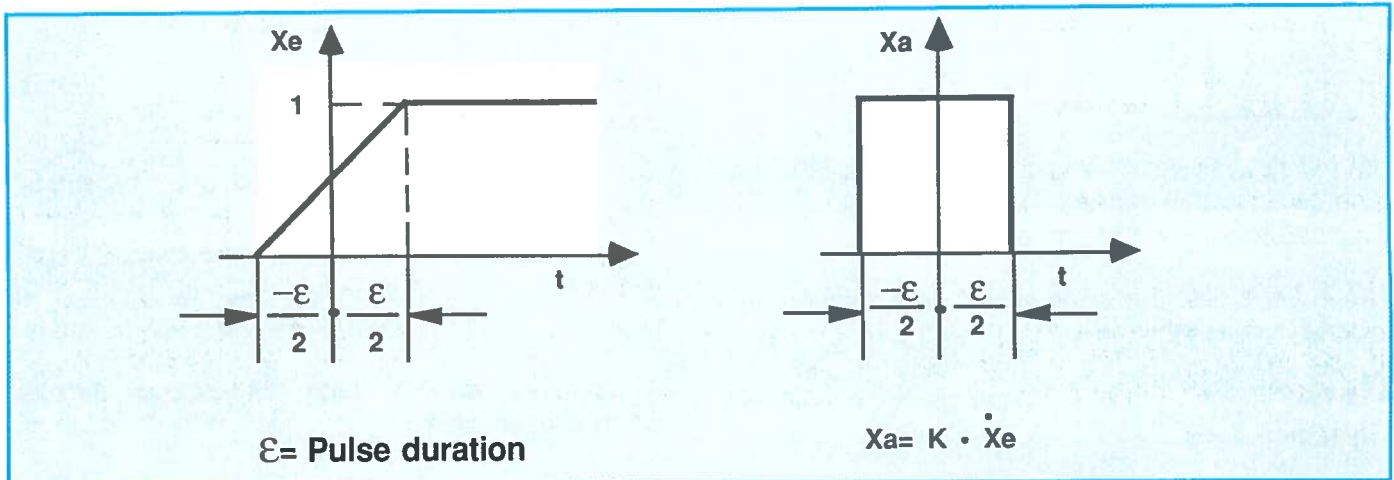


Fig. 22 Stepped response

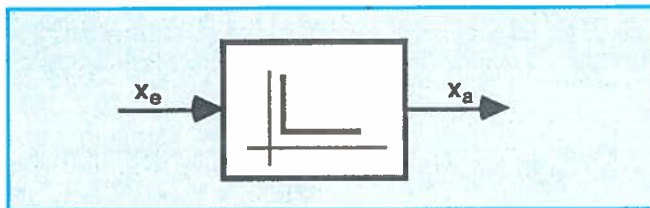


Fig. 23 Symbol of the D-element

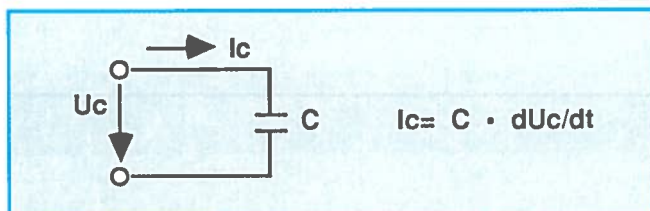


Fig. 24

Examples of the D-element are the relationship  $U = L \cdot I$  of the voltage  $U$  with respect to the input current  $I$  of an inductance, or the charging current  $i$  of a capacitor, dependent on the capacitance  $C$ , and the applied voltage  $U_C$  or the relationship  $F = m \cdot v$  ( $v=a$ ) i.e. the dependence of force  $F$  on velocity  $v$ .

## Dead Time Element

The amount of material at the beginning of belt is  $X_e$ , is equal to the amount discharged at the end of the belt,  $X_a$ . At the time  $t$ , the quantity at the start of the belt is  $X_e(t)$ , the time  $T_t = l/v$  elapses until this quantity is conveyed to the end of the belt.

At time  $t$  therefore, at the end of the belt is the quantity which was earlier at the beginning of the belt at time  $T_t$  i.e. at time  $(t - T_t)$

Therefore

$$X_a(t) = X_e(t - T_1)$$

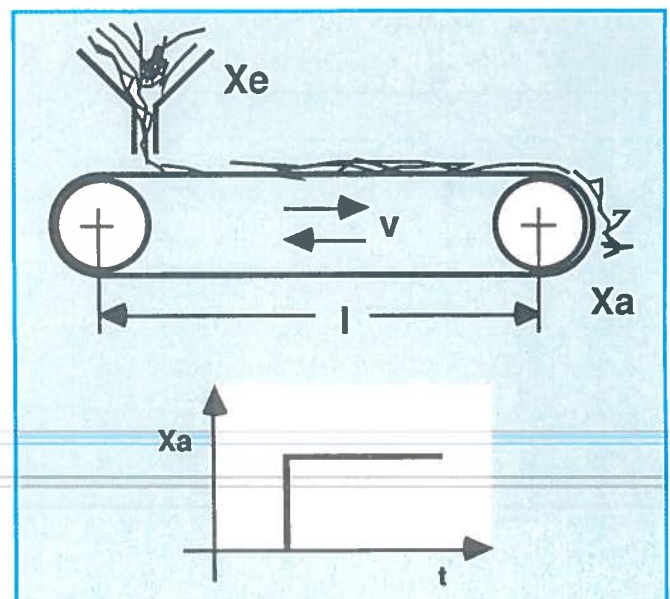


Fig. 25 e.g. Conveyor belt

## Proportional element with 1st order deceleration P - T<sub>1</sub>-element

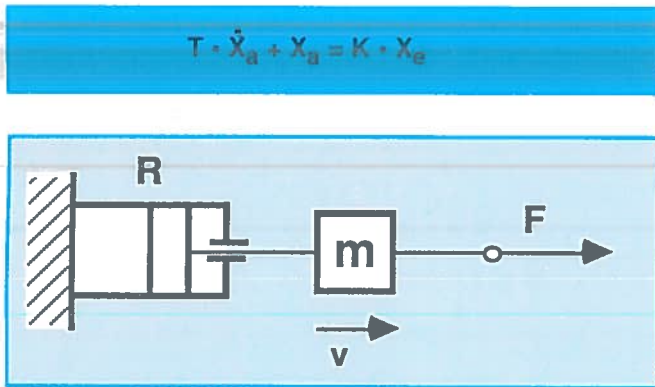


Fig. 26 Example of a P-T<sub>1</sub>-element

The outer forces  $F$  and the velocity-proportional fluid friction  $R \cdot v$  act on the mass  $m$ .

Therefore

$$m \cdot \dot{v} = F - R \cdot v$$

or

$$m/R \cdot \dot{v} + v = F/R$$

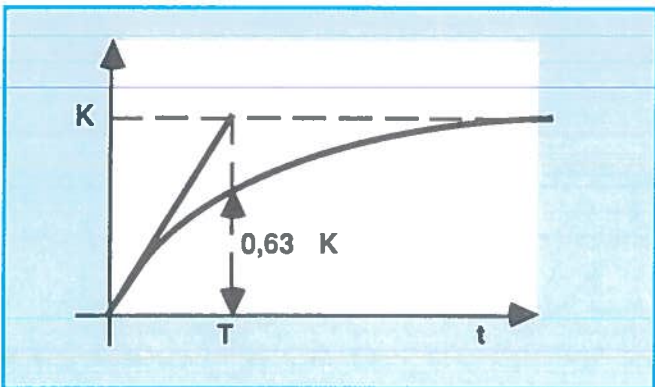


Fig. 27 Response of the P-T<sub>1</sub>-element to stepped input

The final value  $K$  is reached only after a time delay. The dynamic effect of the P-T<sub>1</sub>-element is in a delay of  $x_e(t)$ .

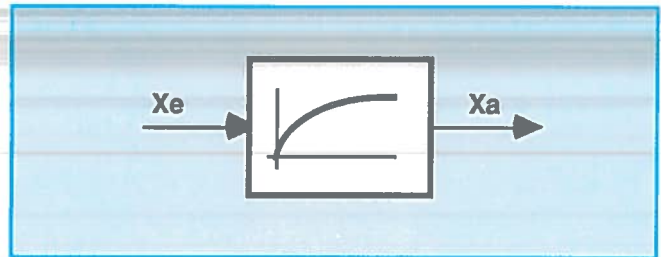


Fig. 28 Symbol for a P-T<sub>1</sub>-element

The tangent for the stepped response at  $t = 0$  assumes the final value  $K$  at the time  $t = T$ .

$T$  is therefore referred to as the "time constant" of the P-T<sub>1</sub>-element.

The time constant  $T$  determines the rate of rise.

## Proportional element with 2nd order deceleration P-T<sub>2</sub>-element

The P-T<sub>2</sub>-element is defined by the equation

$$T^2 \cdot \ddot{x}_a + 2 D T \cdot \dot{x}_a + x_a = K \cdot x_e$$

The constant  $T$  is also termed the time constant, the dimensionless number  $D$  represents the damping, and  $K$  is the transfer constant of the P-T<sub>2</sub>-element.

Relationship between the force  $F$  and the displacement  $X$  of the mechanical system (Fig. 29).

$$m \cdot \ddot{X} = F - R \cdot \dot{X} - C \cdot X$$

or

$$m/c \cdot \ddot{X} + R/c \cdot \dot{X} + X = 1/c \cdot F$$

$T^2$	$2 D T$	$K$
$T = \sqrt{m/c}$	$D = R/(2 \cdot \sqrt{m \cdot c})$	$K = 1/c$

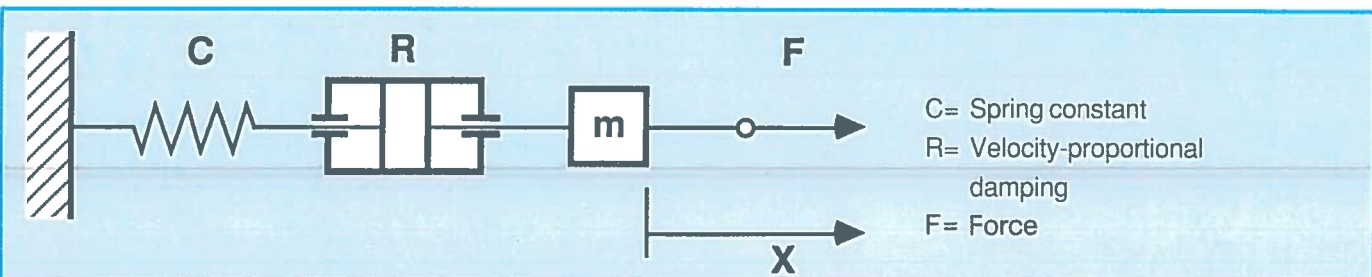


Fig. 29 Example of a P-T<sub>2</sub>-element

### Stepped response of P-T<sub>2</sub>-element

**D > 1**

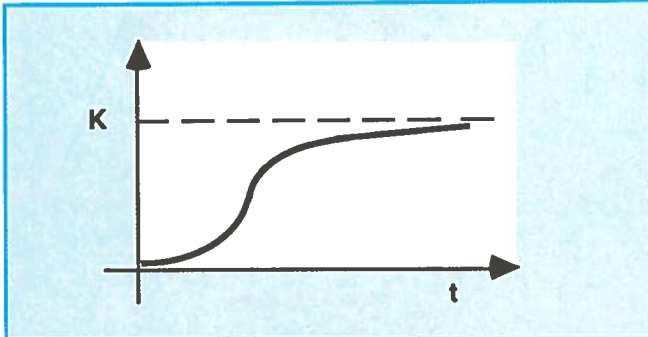


Fig. 30

If  $D > 1$  aperiodic borderline case applies (Fig. 30).

**D < 1**

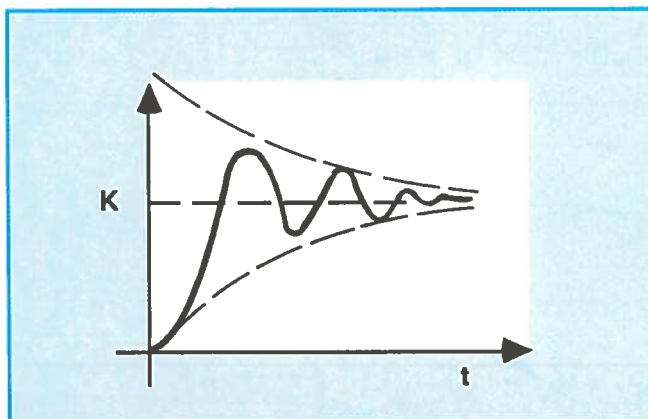


Fig. 31

For  $D < 1$  the stepped response takes the form of a damped oscillation.

Its frequency is

$$\omega_N = \sqrt{1-D_2} \cdot \omega_0 = \sqrt{1-D_2}/T$$

$$\omega_0 = 1/T$$

This is termed the periodical case so that the P-T<sub>2</sub>-element is also referred to as an oscillation element.

The symbol, applicable for all cases, of the P-T<sub>2</sub>-element is derived from this stepped response.

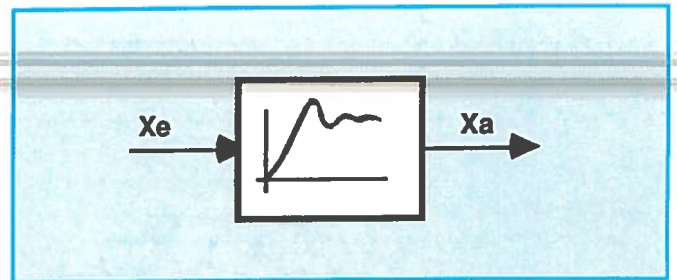


Fig. 32 Symbol for the P-T<sub>2</sub>-element

### Summary of the Elementary Transfer Elements



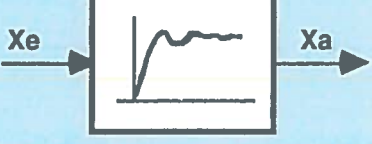



<b>P element</b>	$X_a = K_P \cdot X_e$	
<b>P-T<sub>1</sub> element</b>	$T_1 \cdot \dot{X}_a + X_a = K_P \cdot X_e$	
<b>P-T<sub>2</sub> element</b>	$T_2 \ddot{X}_a + 2 D T \dot{X}_a + X_a = K_P \cdot X_e$	
<b>I element</b>	$X_a = K_I \int X_e(t) dt$	
<b>D element</b>	$X_a = K_D \cdot \dot{X}_e$	
<b>T<sub>t</sub> element</b>	$X_{a(t)} = K \cdot X_e(t - T_t)$	

Fig. 33 Elementary transfer elements

As already mentioned in the introduction, the main task of a closed loop control is to eliminate the effects of the interfering variables on the control variable.

In the case of changes in the signal value, this arrangement is, however, also capable of compensating the actual value of the control variable with respect to the new signal value.

The closed loop control therefore has two tasks:

- Compensation of interfering variables
- Following the control of the input variable.

Influencing the control variable after a change in the input or interfering variable generally requires a certain period of time (refer to transfer function). If the interfering variable undergoes a stepped increase, the

closed loop control responds by readjustment of the control variable. This always takes place with a delay irrespective of the physical nature of the closed loop system.

For example, moment of inertia and friction are of importance in a mechanical system, while signal conversion times (capacitors etc) are present in electrical systems. The progression of the control variable with time is, however of decisive significance for the characteristic of the closed loop control.

For example, the system can begin to oscillate if attempts are made to keep this delay as short as possible is allowing the controller to intervene to a great extent in the case of changes in interfering variables.

If any initial oscillation is self damping, then the closed loop control circuit is considered as being stable. If the oscillation does not abate, i.e. the closed loop control circuit constantly oscillates, then the closed loop control is referred to as being unstable. If the closed loop control is stable, it must also maintain the control difference below a preset value (within a tolerance band).

These requirements with regard to stability and maintaining preset control differences represent absolutely essential requirements which a closed loop control circuit must meet.

Even further demands are often placed on a closed loop control.

For example, the settling time after a change in the input variable, or a disturbance variable, may have to be performed within a certain time.

These requirements are by no means the automatic outcome of adding arbitrary measuring or comparison devices to the machine, and control loop.

The closed loop control circuit would almost certainly, be either unstable, extremely inaccurate or very slow.

To ensure the closed loop control circuit can meet specific requirements, particular fundamentals must be taken into consideration, particularly the selection of the controller itself.

An accurate description of the dynamic characteristic of all elements in a closed loop control circuit is necessary to facilitate correct selection of the controller.

At this point, we will not discuss the many stability criteria but rather we refer to the relevant literature on the subject of closed loop control technology.

Reference is, however, made to general assignment of suitable controllers to given controlled systems.

As shown in Fig. 34, with regard to their time characteristic, it must be possible to adapt the controllers to the relevant controlled system in order to produce stable closed loop control circuits. For this reason, controllers are necessary with varying time characteristics.

Note to Fig. 34:

Control signifies:

Used for change in control variable.

Interference signifies:

Used to compensate interfering variables.







Controlled system	Controller	P	I	PI	PD	PID
 Pure dead time		Not applicable	Slightly poorer than PI	Control + interference	Not applicable	Not applicable
 Dead time + delay 1st order		Not applicable	Poorer than PI	Slightly poorer than PID	Not applicable	Control + interference
 Dead time + delay 2nd order		Not suitable	Poor	Poorer than PID	Poor	Control + interference
 1st order + extremely short idle time (delay time)		Control	Not suitable	Interference	Control at delay time	Interference at delay time
 Higher order		Not suitable	Poorer than PID	Slightly poorer than PID	Not suitable	Control + interference
 Integral characteristic		Control (without delay)	Not applicable unstable structure	Interference (without delay)	Control	Interference

Fig. 34 Selection of a suitable controller for a given controlled system

## Summary of Applicable Controller Functions

The controllers represented in the following can be achieved by the corresponding connection of an operational amplifier.

**P-controller** (proportional closed loop control characteristic)

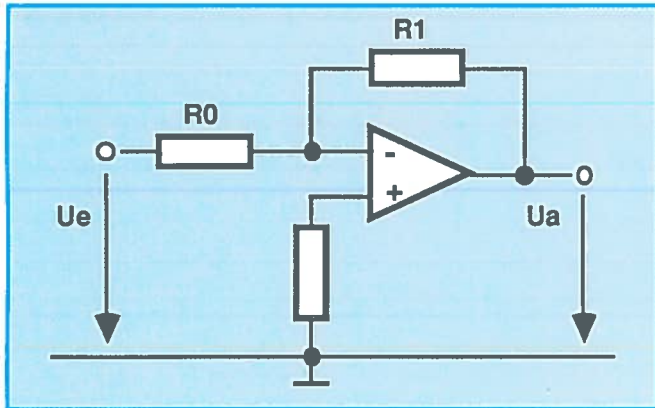


Fig. 35

Proportional control characteristic means that the output value  $U_A$  and input value  $U_E$  are proportional to each other.

The circuit shown above can be represented by the formular

$$U_A = -R_1/R_0 \cdot U_E$$

$R_1/R_0$  = Amplification factor =  $K_P$

The response of a closed loop amplifier to a stepped input is used to determine its characteristic. This refers to the progression with time of the output voltage  $U_A$  when the input voltage  $U_E$  is stepped from zero to a set value.

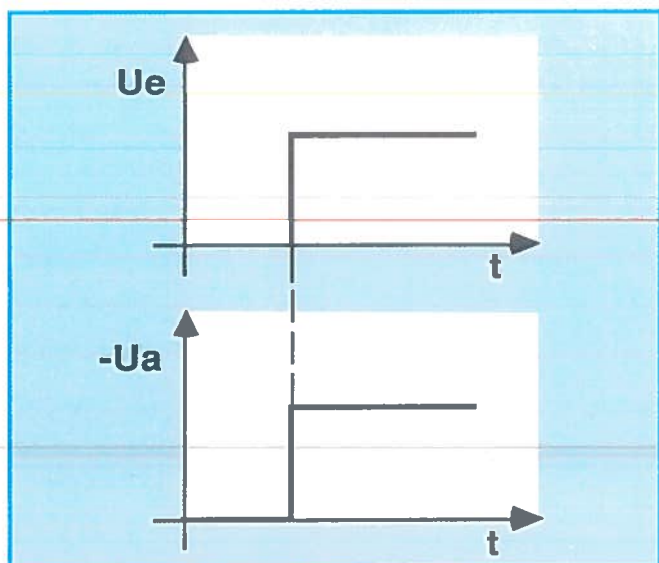


Fig. 36 Step response of P-controller

The P-controller therefore responds to a stepped change in the input variable with a stepped change in the output variable (controlled variable).

### Advantages of the P-controller

- Uncomplicated design
- Easy adjustment
- Fast response to change in control variable

### Disadvantages of the P-controller

Basically, the control variable can never equal the input variable with a P-controller. A remaining closed loop error dependent on the amplification factor must always be accepted.

This is due to the fact that the function of the P-controller requires a closed loop error.

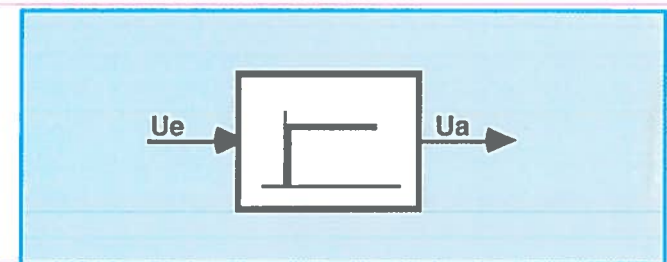


Fig. 37 P-controller represented as a block symbol

# I-Controller

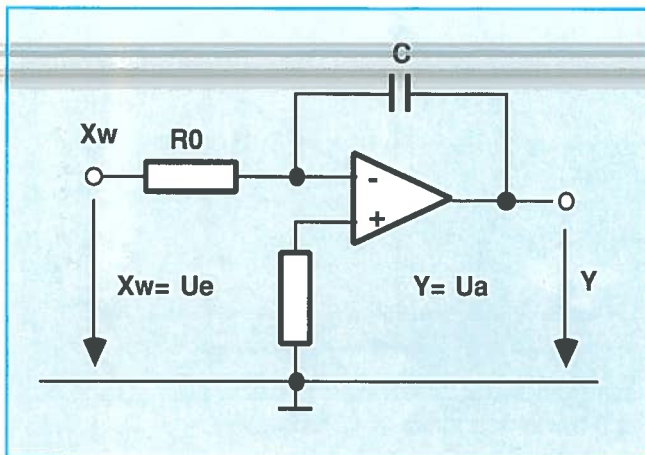


Fig. 38

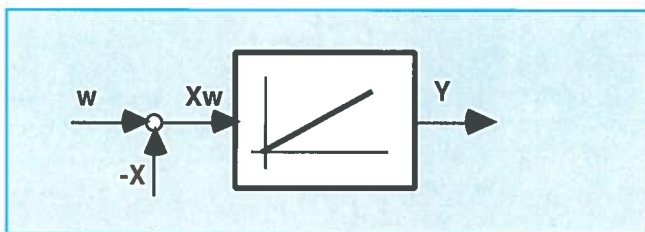


Fig. 39 I-controller represented as a block symbol

A controller with an integrating function produces the time integral of the input variable.

Characterized by the integration time constant

The integral time constant gives the time required by the integrator to achieve an output voltage  $U_a$  with a stepped input voltage  $U_e$  at the input.

$$T_I = R_0 \cdot C$$

or their reciprocal

$$K_I = 1/T_I$$

The input voltage  $U_E$  is the closed loop difference  $w - x = X_w$ .

The output voltage is the positioning variable

$$Y = U_A(t) = -1/T_I \int U_E dt$$

The output signal is inverted if required by the circuit.

A voltage step at the input produces a linear change with respect to time in the output voltage.

A particular feature of an I-element is therefore that the output variable changes for as long as the input variable is not equal to zero. The output voltage remains at the value achieved when the input voltage rests at zero (Fig. 41).

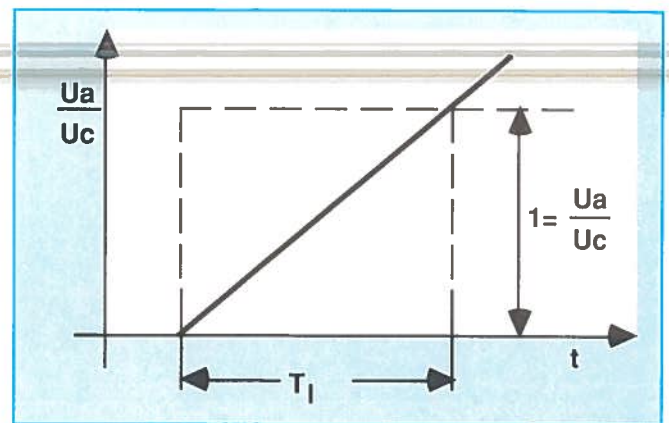


Fig. 40

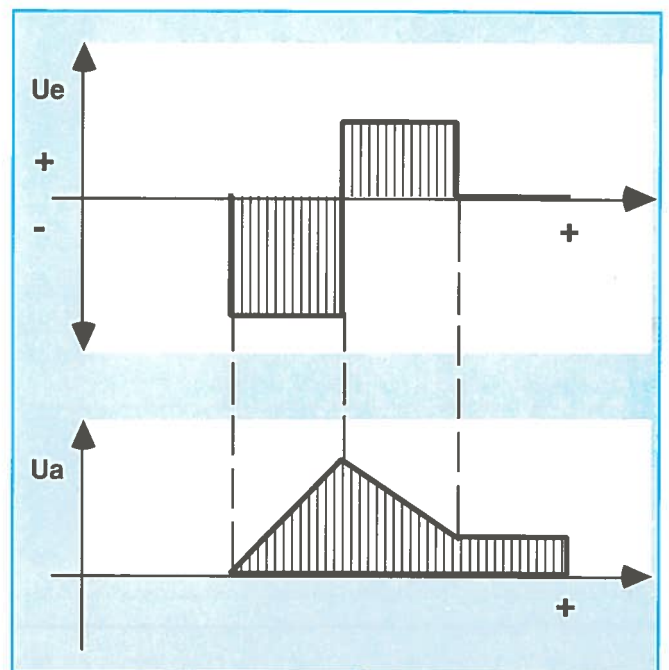


Fig. 41 Time diagram of an I-controller

Compared to the P-controller, the positioning variable generated by the I-controller is not proportional to the closed loop error, but rather the change with time in the positioning variable is proportional to the closed loop error.

In principle, the integral controller completely eliminates all closed loop errors since in time even the smallest input signal increases to become a large output signal.

This advantage, the elimination of closed loop errors, must be offset by certain disadvantages.

As can be seen from the time diagram of the I-controller, the response is relatively slow to a change in the control variable. This results in long positioning times and considerable overshoot of the closed loop control variable can occur.

## PI-Controller

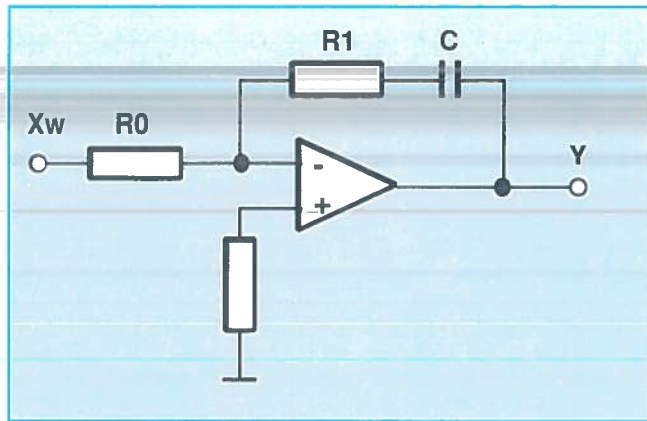


Fig. 42 PI-controller

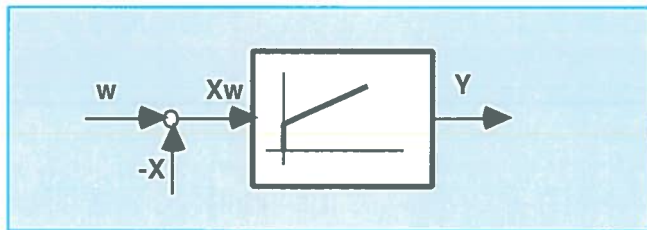


Fig. 43 PI controller represented as block symbol

In the PI-controller (proportional-integral controller), the advantageous characteristics of the P-controller (fast response) are combined with those of the I-controller (accuracy).

$K_P = R_1/R_0$	$T_I = R_0 \cdot C$
$T_n = R_1 \cdot C = K_P/K_I$	$K_I = 1/T_I$

The PI-controller is characterized by the constants  $K_P$  and  $K_I$  as well as the resetting time  $T_n$ .

The delay in setting time is the time ( $T_n$ ), taken by the integral part, to produce the same output as the proportional part (which would be produced instantaneously), for the same change in signal input.

In other words:

The characteristic of the PI-controller, is the same as that of an I-controller, but with its effect "moved forward in time" by the settling time  $T_n$  (Fig. 44).

This controller is mainly used in applications where the proportional share compensates an interfering variable quickly yet not so accurately, while the integral component ensures exact and full correction.

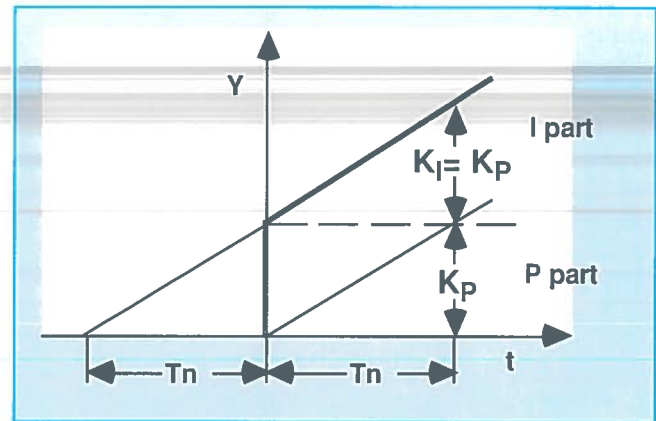


Fig. 44 Characteristics of a PI controller

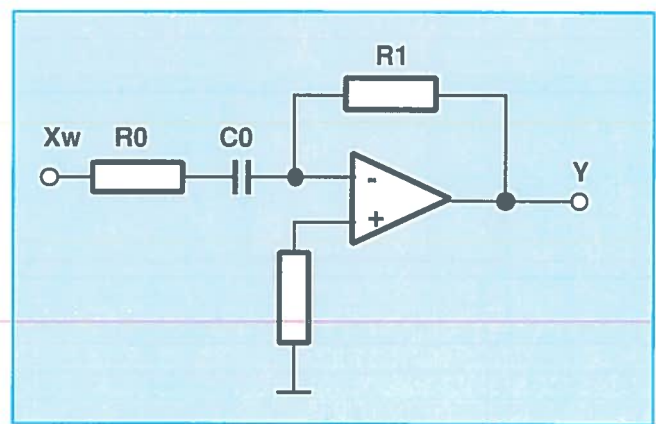


Fig. 45 D controller

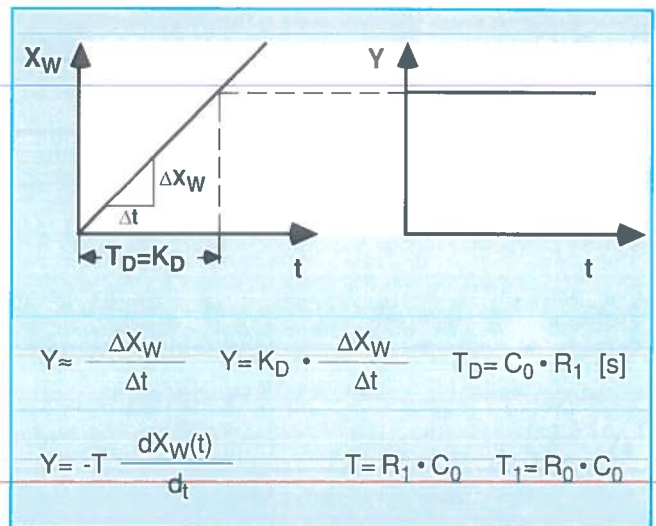


Fig. 45a Characteristics of a D controller

## D-controller

The differential controller responds to the change in speed of the closed loop error ( $\Delta X_W/\Delta t$ ).

This controller is therefore not tested with a stepped input signal but rather with a ramp input signal.

A characteristic feature is the differentiation time constant  $T_D$  or the controller constant  $K_D$ .

Normally, this controller is only used in conjunction with other controllers.

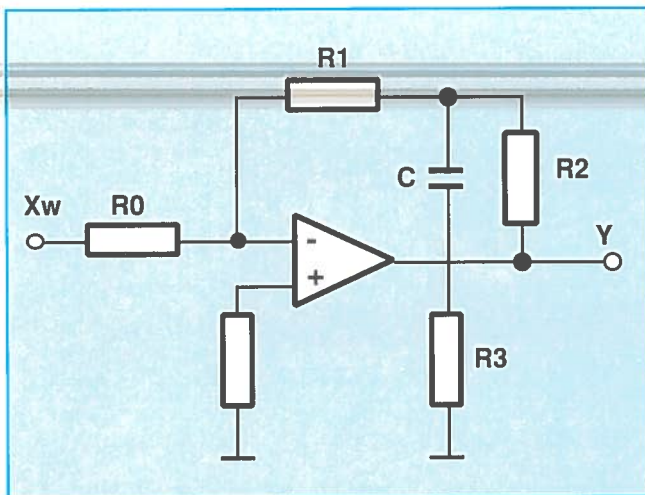
PD-T<sub>1</sub>-Controller


Fig. 46

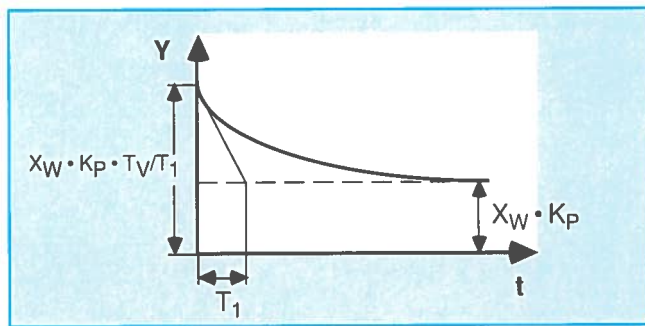


Fig. 47 PD controller represented as block symbol

The time delay  $T_1 = C \cdot R_3$  delays the output signal and limits it at time  $t = 0$  to:

$$X_W \cdot K_P \cdot T_V / T_1.$$

$$(K_P = (R_1 + R_2) / R_0; \quad T_V = [R_1 \cdot R_2 / (R_1 + R_2)] \cdot C)$$

If a PD controller without a time delay is tested with a ramp function input, the advance time  $T_V$  can be determined.

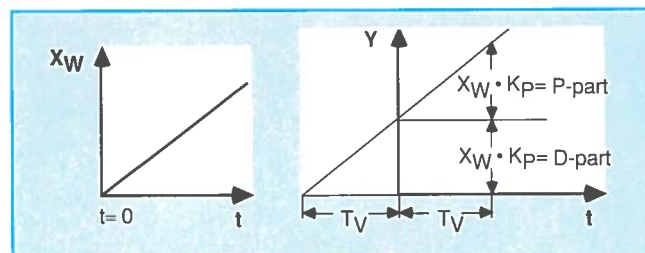


Fig. 47a Characteristics of a PD controller

The advance time  $T_V$ , is the time required by the P part, to achieve the value of output signal of the D part at the time point  $t = 0$ .

A differential stage in the proportional controller accelerates the control-process since the rate of change in the closed loop error also influences the output signal.

However, the PD-controller does have a static closed loop error.

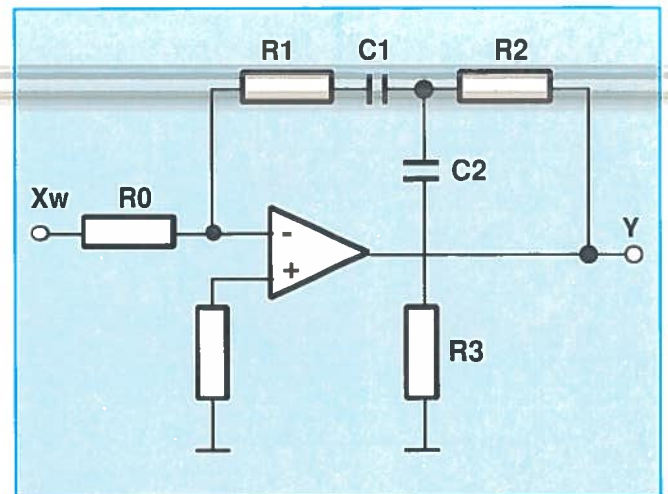
 PID-T<sub>1</sub>-Controller


Fig. 48

The PID-controller represents a combination of all three types of controllers.

Added to the good dynamic properties of the PD-controller, is the fact that the static closed loop error disappears.

Such a controller with variable controller constants can be adapted to any controlled system.

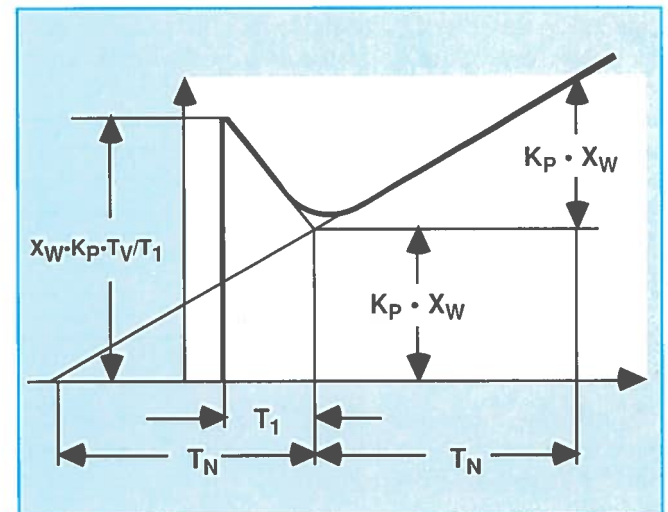


Fig. 49

Initially, positioning variable changes by an amount dependent on the change in velocity of the input variable  $dx_d/dt$  (D-component). After the approach time has elapsed, the positioning variable returns to a value corresponding to the proportional range and then changes corresponding to the value of the I-component.

$$K_P = (R_1 + R_2) / R_0$$

$$T_N = R_1 \cdot C_1$$

$$T_V = R_2 \cdot C_2$$

$$T_1 = R_3 \cdot C_2 \quad \rightarrow \text{damping time constant}$$

$$R_3 = \text{Damping resistance (see PD-controller)}$$

## Closed Loop Position Control, Motor Drive

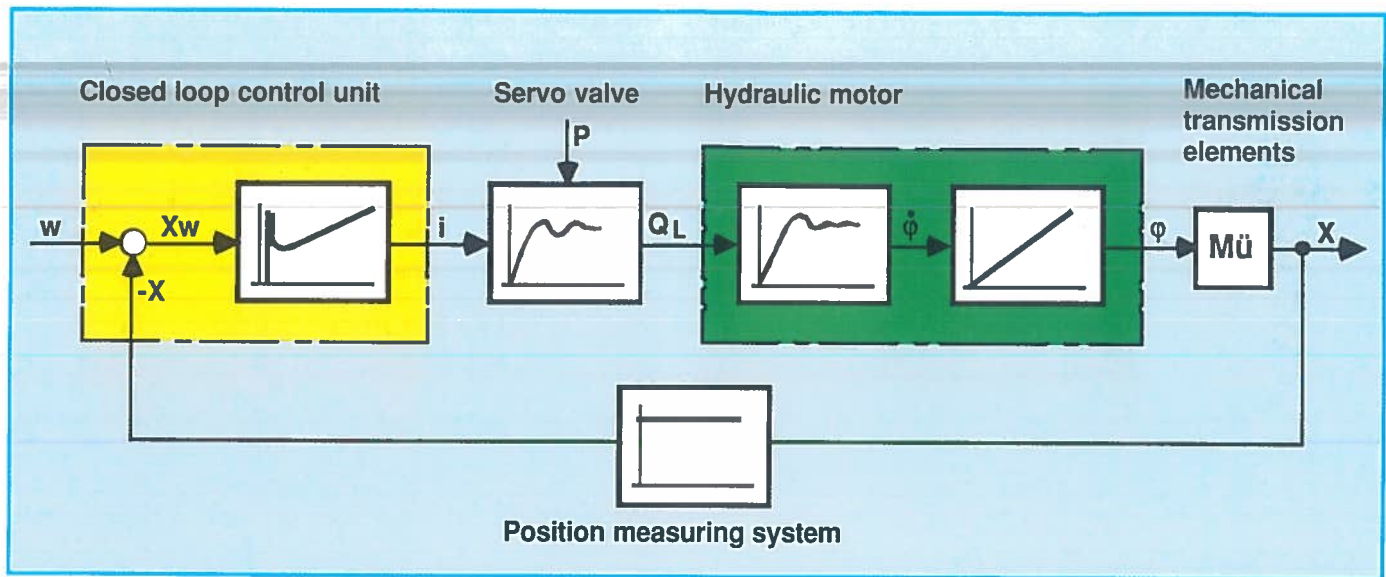


Fig. 50 Block diagram of a position-controlled motor drive

### Transmission Characteristic of the Individual Closed Loop Control Elements

The servo valve and the loaded servo motor are considered as 2nd order systems connected in series (proportional element with 2nd order delay,  $PT_2$ -elements).

Due to the integration during the transition from angular velocity to angle of rotation, the controlled system is considered as a 5th order system (by means of multiplication of the frequency response equations of the individual elements. Refer to relevant literature on closed loop control technology)

The controller which is selected, is a PID-controller corresponding to the selection criteria of a controlled system Fig. 34.

The position measuring system is considered as a P-element without delay, i.e. it responds to a change in the input variable without delay.

The hydraulic motor has proportional transition characteristics referred to the angular velocity and integral characteristics referred to the angle of rotation.

## Closed Loop Position Control, Cylinder Drive

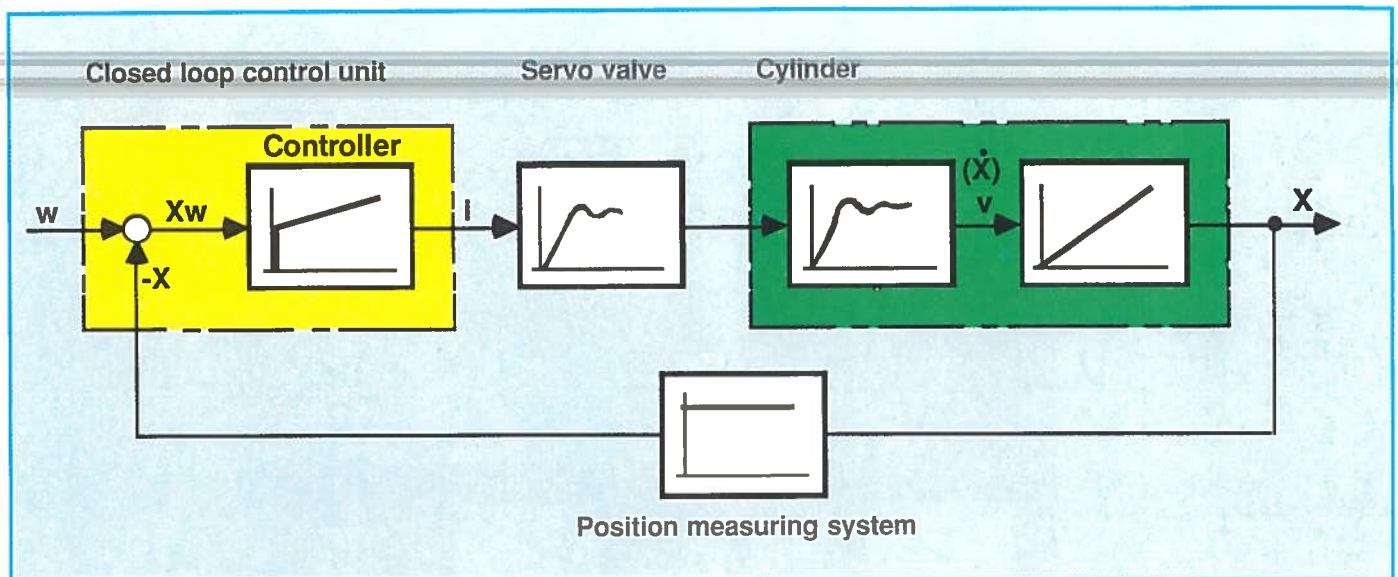


Fig. 51 Block diagram of a position-controlled cylinder drive

### Transmission Characteristics of the Individual Closed Loop Control Elements

The servo valve and the cylinder are once again considered as 2nd order systems connected in series.

In this case, the integration represents the transition from the velocity of the cylinder to the stroke.

Also in this case, a 5th order system results, described in more detail on *Page J4*.

It can be seen that the two block diagrams are almost identical. This therefore confirms the statement made on *Page J6* that when changing from the real technical system to the theoretical model, the differences disappear.

The hydraulic cylinder shows a proportional transition characteristic referred to the travel speed and an integral transition characteristic referred to the cylinder stroke.

## Closed Loop Positional Control (Closed Loop Following Control Circuit)

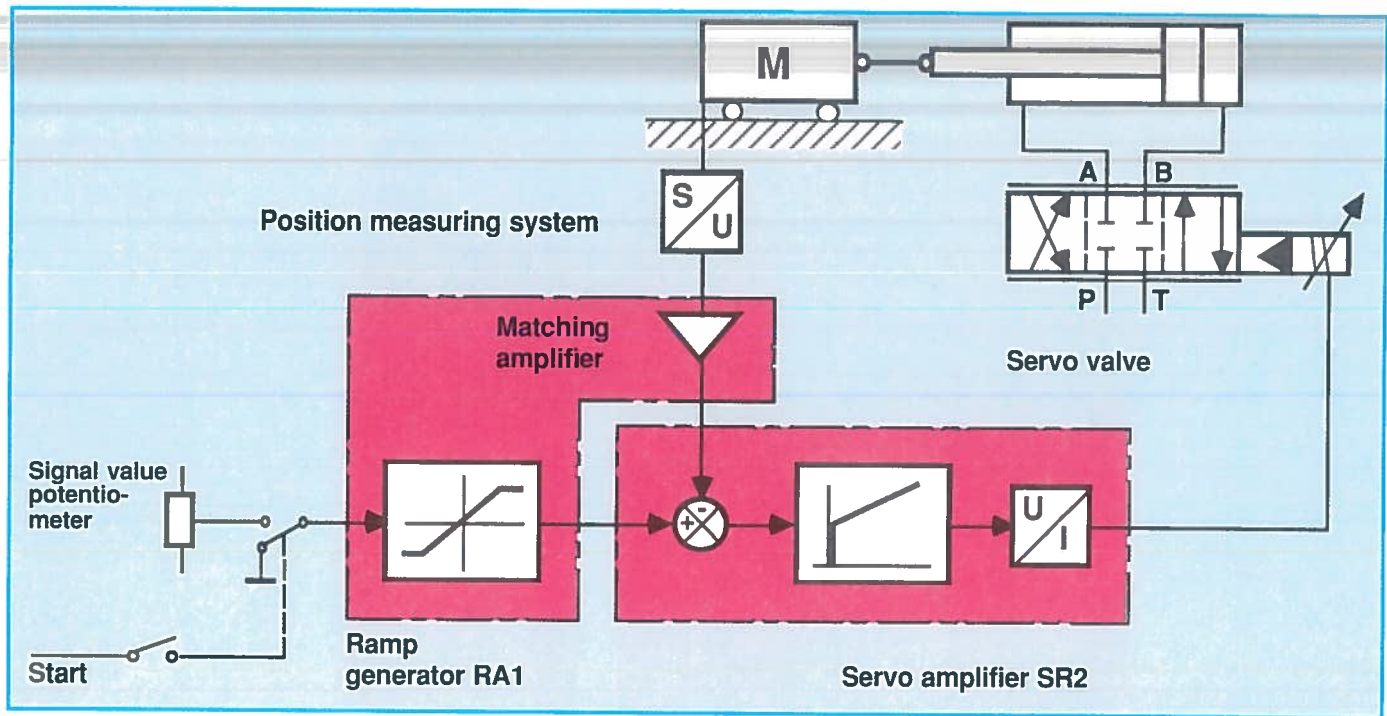


Fig. 52 Example of a closed loop position control circuit

Both the position of the cylinder and the travel speed can be controlled with this circuit.

### Signal Flow

A start signal switches the position signal value to the ramp generator. Over the set ramp time, the output signal of the ramp generator increases from 0 Volt to the voltage value set at the signal value potentiometer.

The set ramp time therefore corresponds to the signal processing speed.

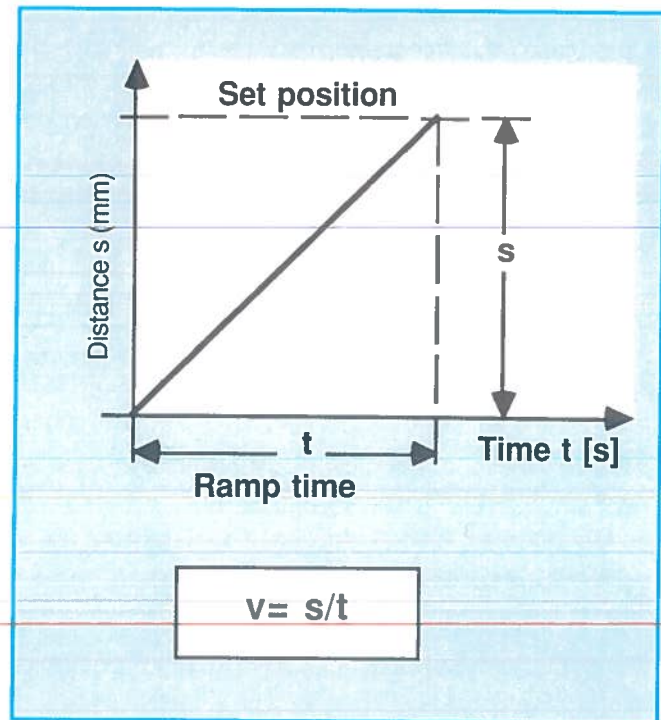


Fig. 53

### Closed Loop Speed Control (Closed Loop Velocity Control) with Integration of Interfering Variables

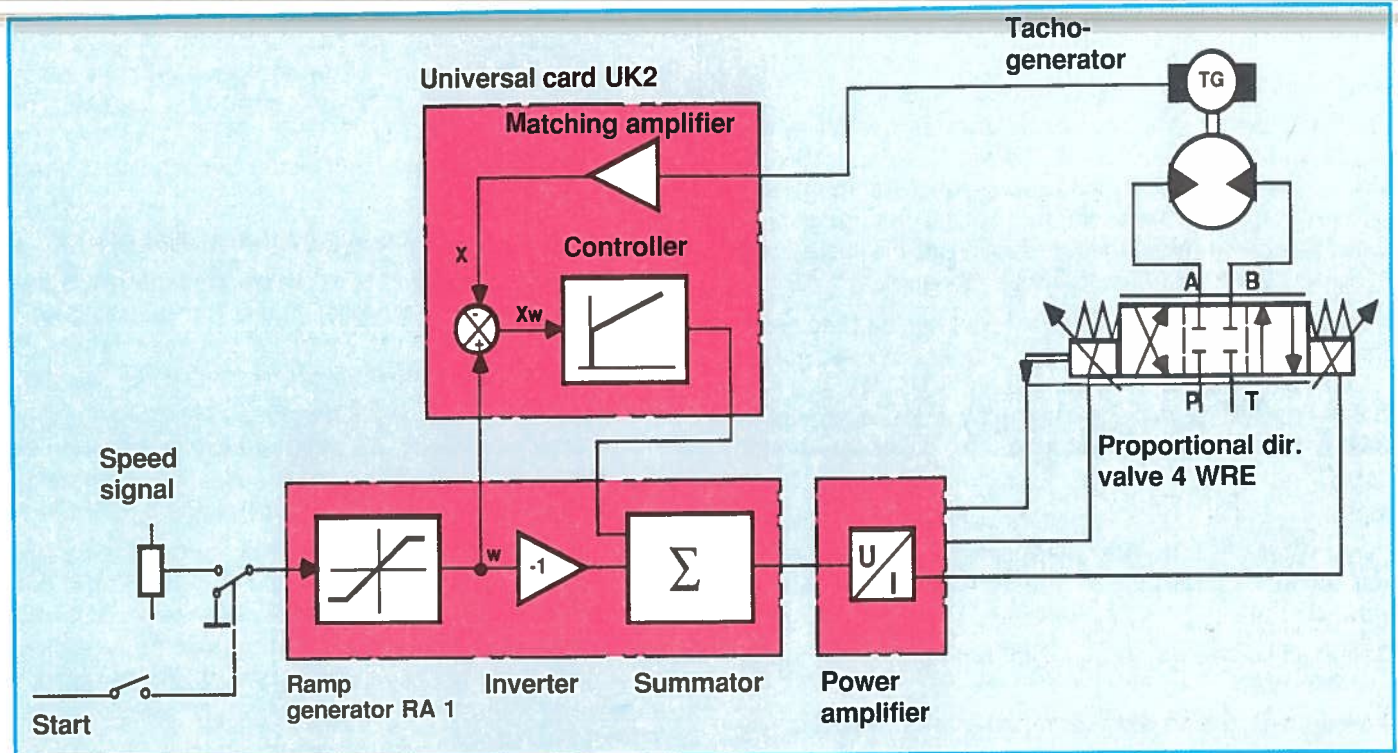


Fig. 54 Example of a closed loop speed control circuit

The start signal switches the set speed signal to the ramp generator.

The signal value at the output of the ramp generator increases corresponding to the set ramp time.

This signal is routed directly to the power amplifier via the inverter and summator so that the valve is directly controlled by this signal value. At the same time, the signal value is compared with the control variable (momentary actual speed) and the difference is fed to the actual controller.

The positioning signal of the controller is fed to the summator where it influences the positioning signal which is routed to the power amplifier and in turn to the servo valve.

This circuit can result in higher dynamics in the closed loop control since the actual controller need only be active in the case of a signal/feedback difference.

### Closed Loop Velocity Control

Only closed loop velocity control is possible with the system shown in Fig. 56.

A definitive position cannot be assumed.

#### Signal Sequence

The start signal switches the velocity signal set at the signal value potentiometer to the ramp generator RA to act as the input signal. The ramp generator increases its output signal corresponding to the set ramp time from 0 V to the signal value applied at the input. The ramp time is the measure for acceleration.

The output signal of the ramp generator is directed to the servo amplifier. The velocity of the cylinder is recorded by means of a velocity sensor. The velocity signal is adapted to the signal value by a matching amplifier, which is also contained on the ramp generator card.

The input signal is normally in the range 0 - 10 V. Adaptation of the actual value therefore means that the actual value signal is also 10 V at the maximum required velocity.

This matched actual value signal is also directed to the servo amplifier.

Signal value/actual value comparison takes place in the servo amplifier. The closed loop difference  $x_w$  is routed to the PI-controller which generates the positioning signal  $y$  to directly actuate the servo valve such that the actual velocity is regulated to the set velocity.

The PI-controller changes its output voltage until the set value/actual value difference is zero. (See PI-controller, Page H16).

To prevent the PI-controller from drifting or to ensure the capacitor is not charged during the start, the

controller is "cleared" by the start signal via a switching amplifier.

The PI-controller has its normal closed loop control function when the relay d1 is energized. On the other hand, if the relay d1 is de-energized, the feedback of the operational amplifier is short-circuited so that the output signal  $y$  equals zero (since the amplification is equal to zero).

Clearing therefore means energization of relay d1.

The controller is cleared by a switching amplifier (1) dependent on the applied signal value. The switching amplifier is set such that the controller assumes its closed loop control function at a signal value of approx. 100mV.

As an additional circuit, a second switching amplifier (2) is shown here which influences the clearing of the controller depending on the actual velocity value.

If, for example, the velocity signal is stepped to zero, the switching amplifier coupled to the signal value de-energizes. The control function is, however, retained by the second switching amplifier coupled to the actual value, so that the movement of the cylinder corresponding to the control characteristic can be continued to the null condition.

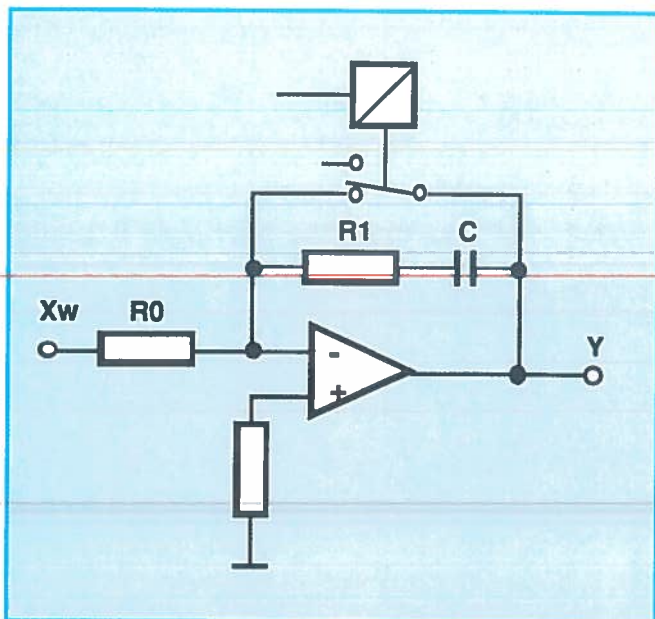


Fig. 55 Circuit for "clearing" the PI regulator

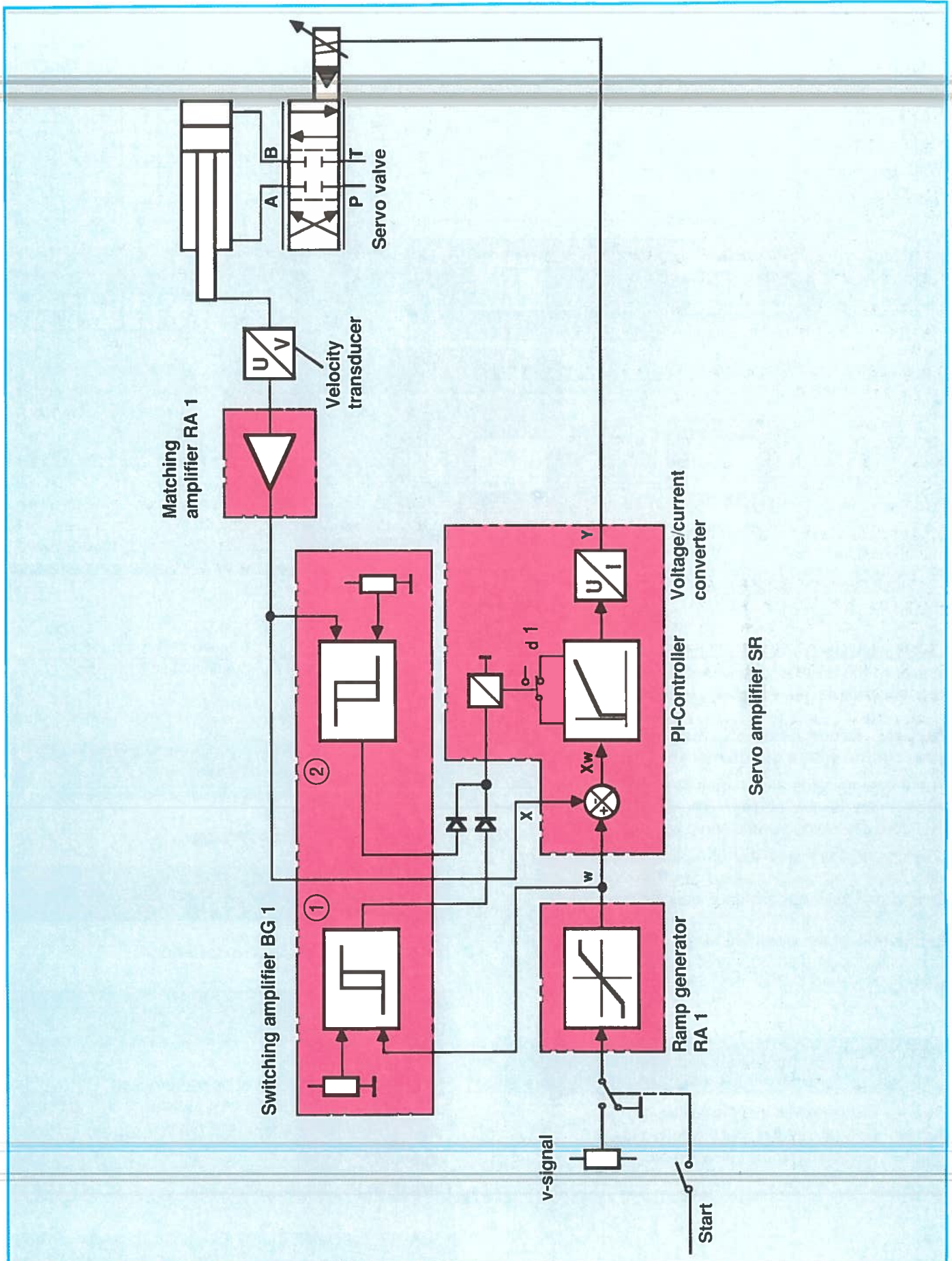


Fig. 56 Example of a closed loop velocity control circuit

## Closed Loop Pressure Control

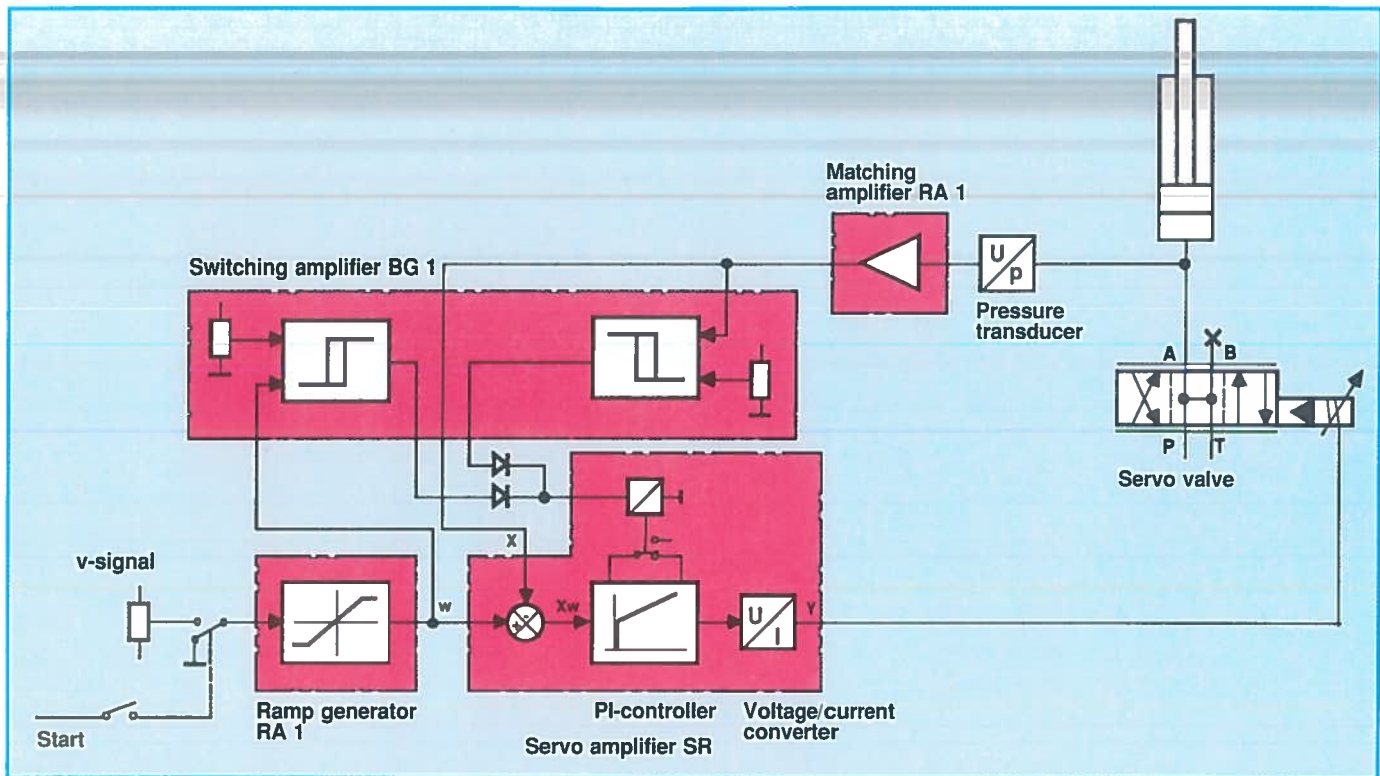


Fig. 57 Example of a closed loop pressure control circuit

The block diagram of the closed loop control circuit is similar to the previous block diagrams so the the signal sequence will not be described in this case.

### General information on the closed-loop pressure control with a directional servo valve

The valve operates about its neutral point provided disturbing flow is not required. The possible loop gain is therefore determined by the servo valve.

The characteristics of the oil volume under compression also has an influence on the system, which means, that the time constant  $T$  must be considered.

### Estimation of the possible loop gain

The critical loop gain is almost proportional to the product

$$V_{crit} = 2 D_V \cdot \omega_V \cdot T$$

- $D_V$  = Damping factor of the valve
- $\omega_V$  = Natural frequency of the valve (1/s)  
(frequency at  $-90^\circ$  phase offset)
- $T$  = Time constant of the pressure volume

If the amplitude drop of the valve at the natural frequency ( $-90^\circ$ ) is defined as  $A_V$  (specified in dB (decibel)) then

$$A_V = 20 \cdot \log (1/2 D_V)$$

resulting in the degree of damping  $D_V$

$$D_V = 10^{-(A_V/20)/2}$$

The time constant  $T$  can be expressed as:

$$T = V/E K_{pq}$$

- $V$  = Oil volume which is to be compressed
- $E$  = Modulus of elasticity of oil ( $1.4 \cdot 10^7$ ) [N/cm<sup>2</sup>]
- $K_{pq}$  = Pressure/flow gain of the valve
- $K_{pq} = Q_{max} / P_{max}$  [cm<sup>3</sup>/s/bar]

The optimum loop gain is therefore

$$V_{opt} = 1/3 V_{crit}$$

### Equipment for Implementing the Closed Loop Control Circuit

Universal electronic cards have been developed to achieve the various closed loop control circuits by relatively simple means.

Any analogue closed loop control circuit can be produced by correctly linking these pc boards.

This is also indicated in the individual examples of closed-loop control circuits in the block diagrams

Fig. 52 Closed loop positional control circuit

Fig. 54 Closed loop speed control circuit

Fig. 56 Closed loop velocity control circuit

Fig. 57 Closed loop pressure control circuit.

## 1 Servo Amplifier

Servo amplifiers are used to actuate servo valves or proportional valves with servo valve pilot control.

Its primary function is to amplify an analogue input signal (signal value, positioning variable) such that the servo valve can be actuated with the output signal (e.g. amplification 1 mA : 60 mA).

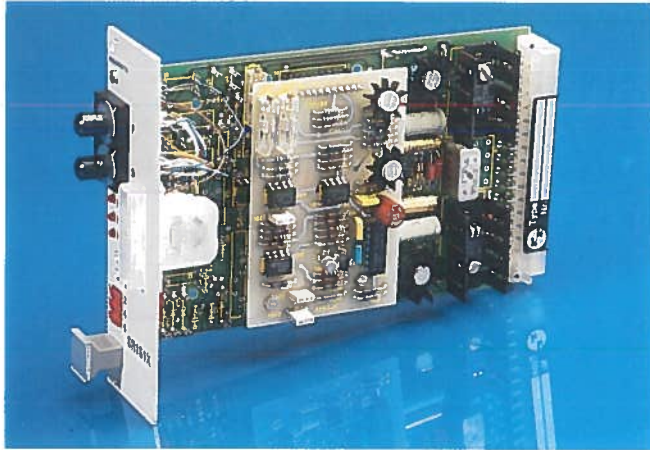


Fig. 58 Servo amplifier Type SR 1

Amplifiers are separated into application areas.

### Servo amplifier SR1

For servo valves or proportional valves with a servo valve acting as the pilot and electrical positional feedback of the main spool. The output current is  $I_{\max} \pm 60 \text{ mA}$ .

### Servo amplifier SR2

For servo valves without electrical feedback. The output current is  $I_{\max} \pm 60 \text{ mA}$ .

The valves are equipped with the flapper jet system which utilises the maximum output current of  $\pm 60 \text{ mA}$ .

### Servo amplifier SR3

For servo valves or proportional valves with a servo valve acting as the pilot and electrical positional feedback of the main spool. The output current is  $I_{\max} \pm 700 \text{ mA}$ .

### Servo amplifier SR4

For servo valves without electrical feedback. The output current is  $I_{\max} \pm 700 \text{ mA}$ .

These two amplifiers with  $I_{\max} \pm 700 \text{ mA}$  for actuation of a control valve are used in conjunction with a single stage torque motor for direct operation of a valve spool.

The design of the servo amplifier is shown in the block diagram (Fig. 60).

A smoothed DC voltage (1) of  $\pm(20 \text{ to } 28) \text{ V}$  is necessary as the supply.

The power supply unit NE 1 S 30 can be used for this purpose. The output voltage is  $\pm 22 \text{ to } 30 \text{ V}$  smoothed, the supply voltage  $220 \text{ V}/50\text{-}60 \text{ Hz}$  or  $110 \text{ V}/50\text{-}60 \text{ Hz}$ .

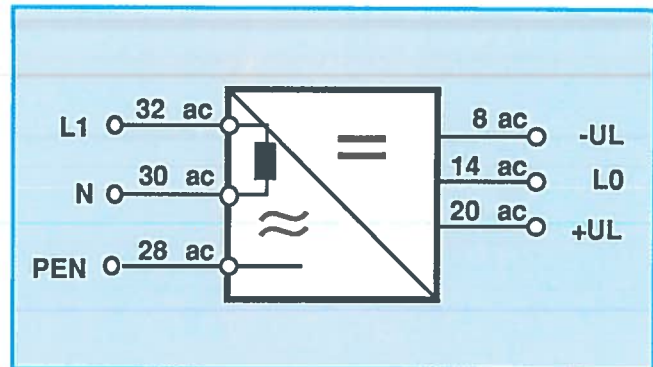


Fig. 59 Power supply unit Type NE 1 S 30

A stabilized voltage (2) of  $\pm 15 \text{ V}$  is then produced from the supply voltage on the amplifier card itself.

This voltage is used for:

- the supply of external consumers such as potentiometers (tapping point at 12c (+15 V) and at 22c (-15 V)

as well as

- the supply of the internal operational amplifiers.

Two basic function groups should be considered

a) the control for the servo valve with the output stage (4) and the PD-controller (3).

In the version without electrical feedback (SR2 and SR4), the signal value is routed directly to the PD-controller (3).

If the card is used for valves with electrical feedback (5), then the PD-controller is used for the closed loop positional control circuit of the valve itself. The inductive positional transducer signals the position of the valve spool. An oscillator/demodulator (5) provides the AC voltage supply and also re-converts the signal feedback. The positional transducer produces an AC signal varying in amplitude, dependent on the position of the valve spool. The demodulator (5) converts this AC signal into a corresponding DC signal. (See page D7).

The position controller (3) of the valve now compares the signal value at 28a (optionally at 30a) with the signal feedback from the valve spool (measurable at test socket (2) or terminal 32a). Depending on the difference between the signal value and feedback the output stage (4) receives from the controller (3) a corresponding signal which it converts into a proportional valve current.

The signal from the output stage (4) can for example, be switched to contacts at (7) and relay K2 dependent on the system pressure. This is of practical use to prevent destruction of the flapper jet system in the servo valve.

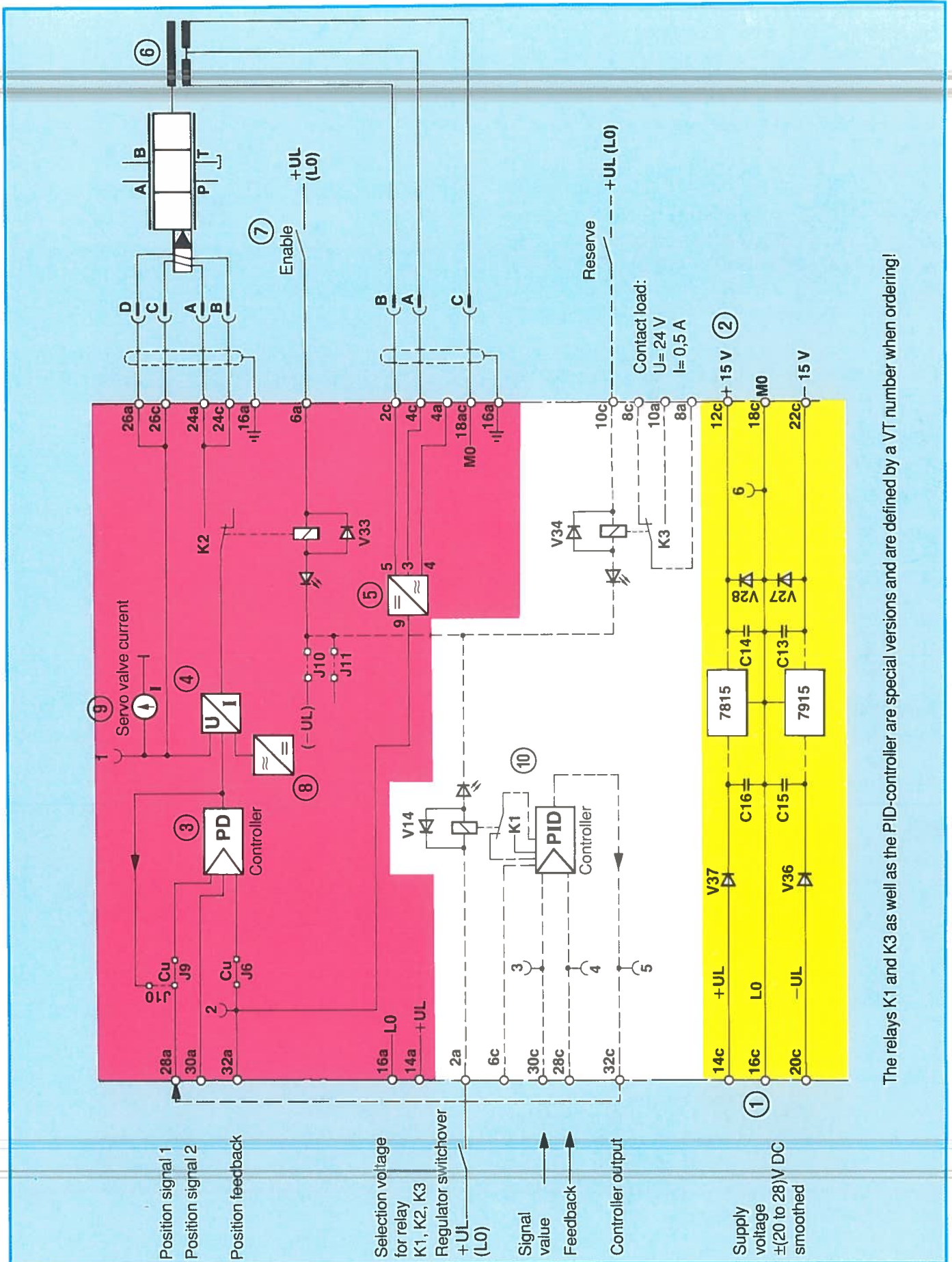


Fig. 60 Block diagram of the servo amplifier Type SR 1 S 30

There is a risk of serious damage to the flapper jet system when the servo valve is actuated without system pressure being applied. For this reason, it is good practice to enable the servo valve by means of a pressure switch in the hydraulic system via input 6a.

Other system-dependent connections can be linked to this input.

A constant dither current (20 mA<sub>SS</sub>) at constant frequency (480 Hz) and amplitude is superimposed upon the valve current by the oscillator (8).

In this way, the hysteresis is decreased and the stability as well as the response sensitivity of the valve increased.

The measuring instrument (9) on the front panel of the amplifier indicates the valve current.

**b)** A second controller (PID) (10) for a superimposed closed loop control circuit. The pc boards can be equipped in this manner on request. By suitable circuitry, the control characteristic is achieved as required by the control functions.

Functional sequence in short: The PID-controller (10) compares the signal value applied at 30c (e.g. velocity signal) with the actual value (feedback) applied at 28c (e.g. velocity signal). Depending on the difference, the controller (at 32a) produces a corresponding signal voltage. This signal must now be directed via 28a of the control for the servo valve.

K1 is used to switch clear the controller (10) selected at terminal 2a.

### 2 Universal Board (UK2)

This board is used to build up any operational circuits. It is equipped with 3 double operational amplifiers and 5 potentiometers, for setting the various neutral points.

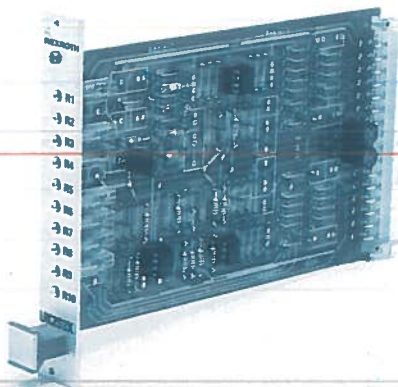


Fig. 61 Universal board UK2

The following functions can be produced:

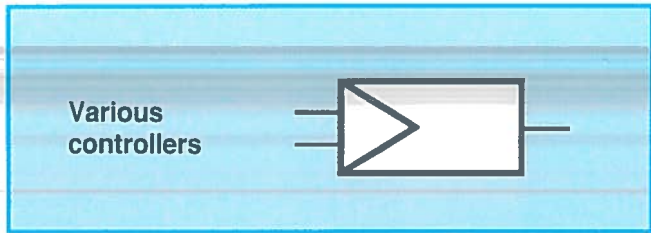


Fig. 62

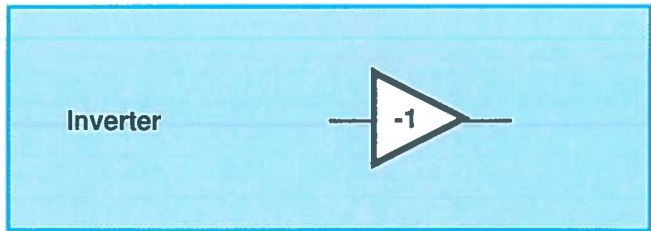


Fig. 63

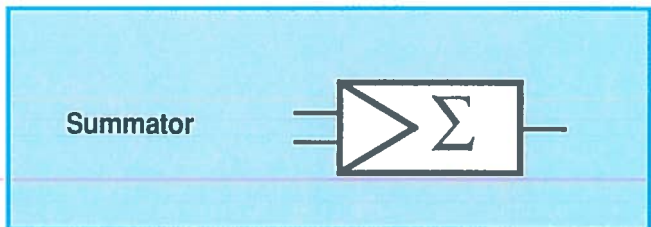


Fig. 64

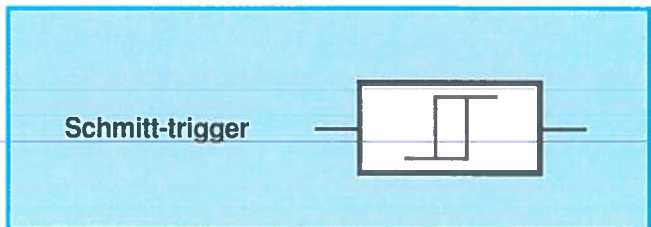


Fig. 65

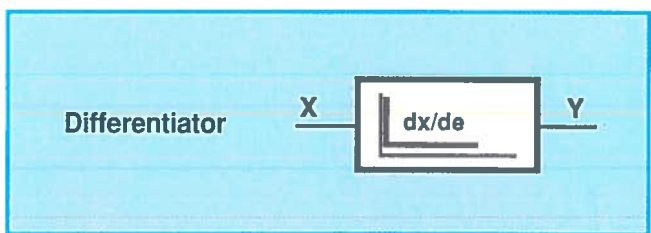


Fig. 66

Circuitry to suit individual requirements is achieved by soldered bridges etc.

The power supply of the card and therefore of the 6 operational amplifiers (3 x double operational amplifiers) must be provided by a stabilized voltage of  $\pm 15$  V.

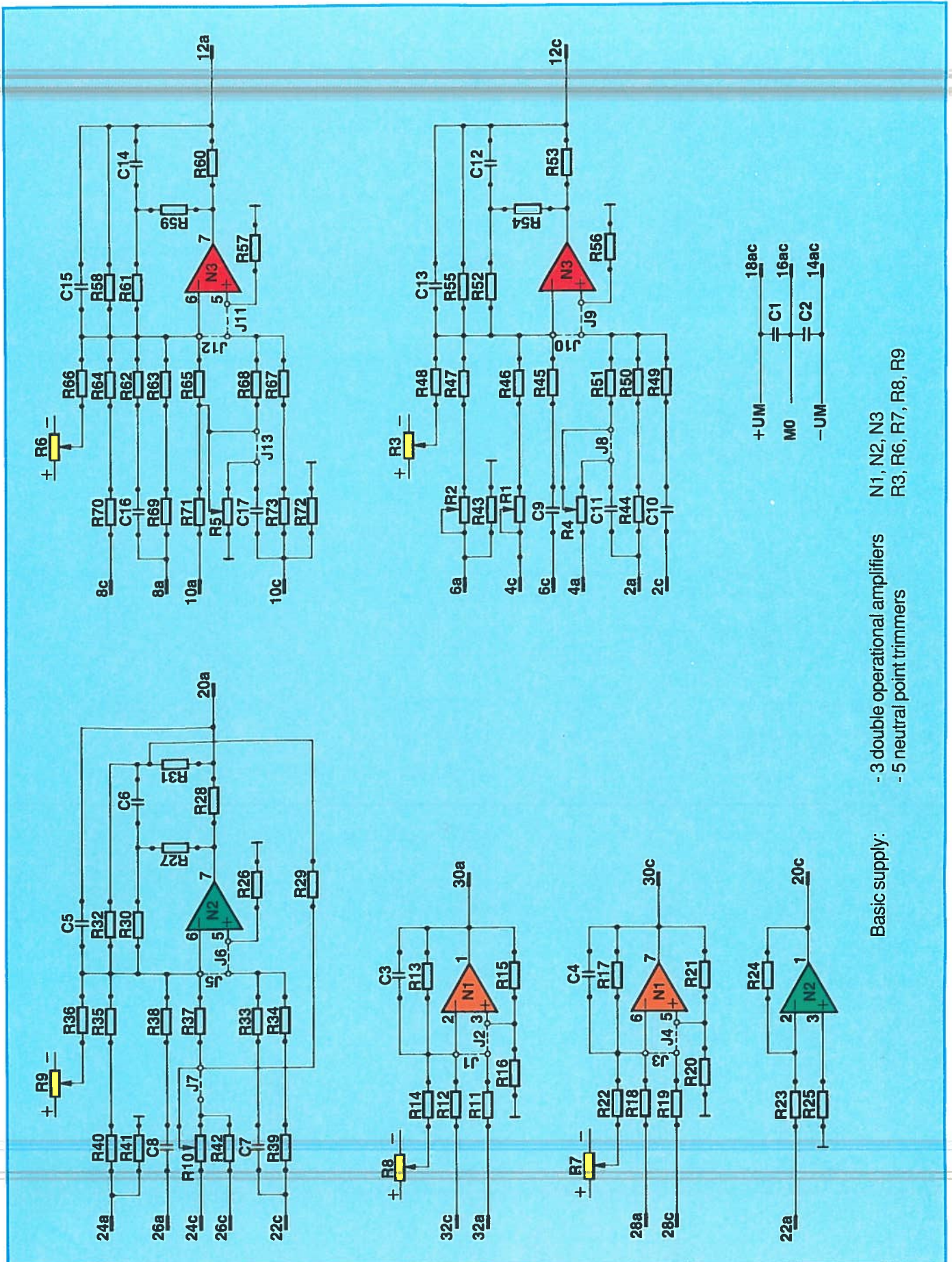


Fig. 67 Terminal connections of the universal card UK 2 S 30

### 3 Card with Ramp Generator RA 1

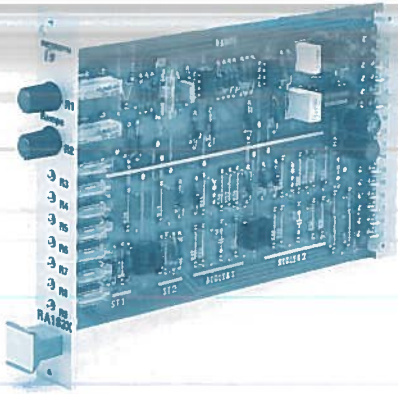


Fig. 68 Ramp generator Type RA 1 S 30

The basic component fabricated on this card is an analogue ramp generator. Corresponding to requirements, one of the three ramp time ranges:

0.01 - 0.1 s

0.1 - 1 s

1 - 10 s

can be selected at a voltage change of 10 V. The ramp times for "up" and "down" can be individually adjusted at the potentiometers P1 and P2. The ramp time can also be set via external potentiometers.

In addition to this ramp generator, the card also contains 5 further operational amplifiers for use as required can be used to realize:

2 controllers (P, PI or PID)

1 inverter

2 switching amplifiers with individually adjustable switching points.

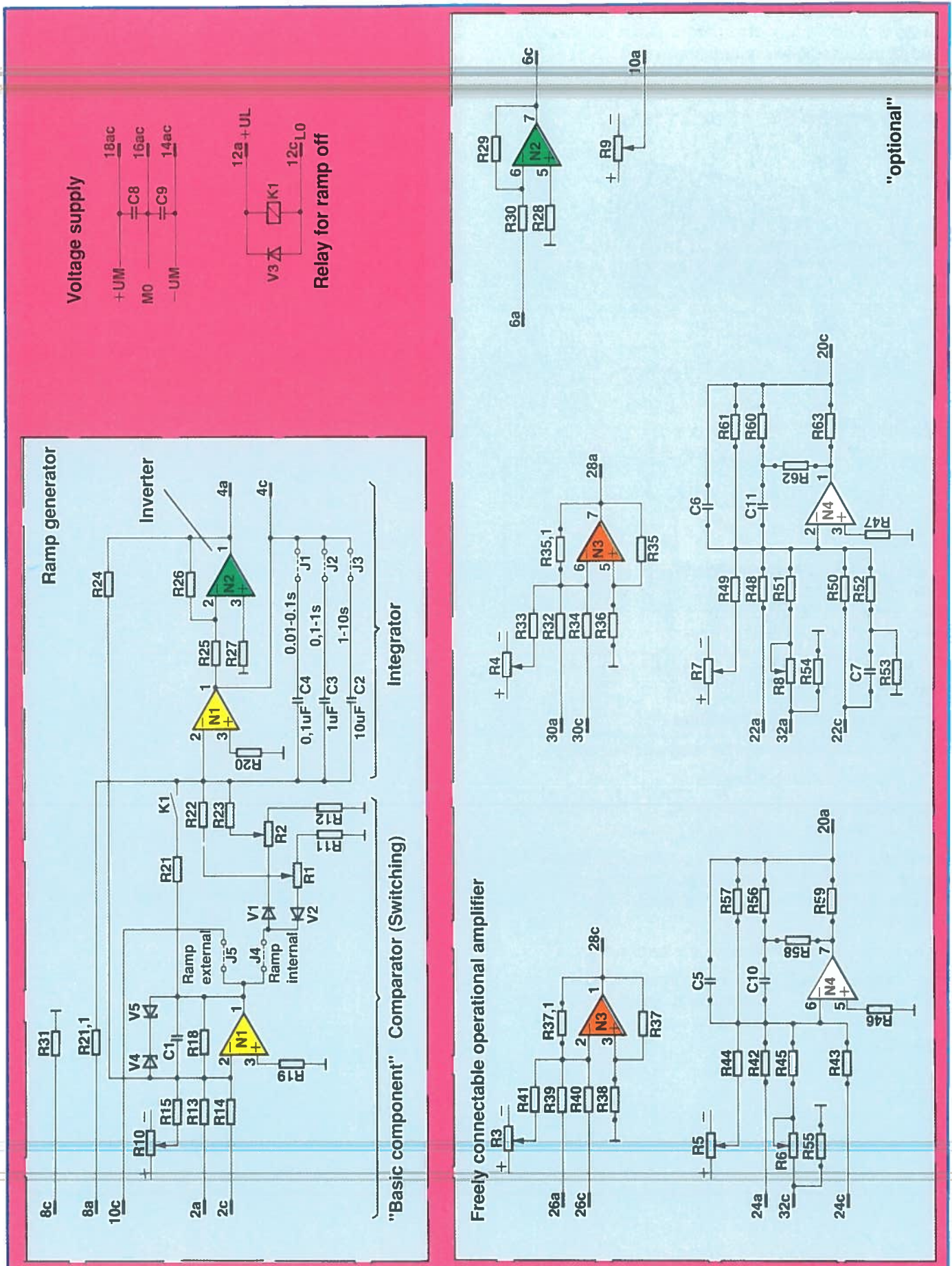


Fig. 69 Terminal connections of the ramp generator RA 1 S 30

In addition to the cards already described, there is of course a wide variety of standard cards still available used for processing analogue signals.

### Limiting amplifier BG 1

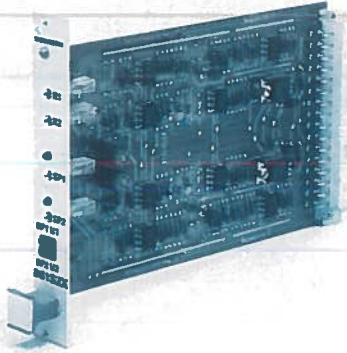


Fig. 70 Limiter Type BG 1 S 30

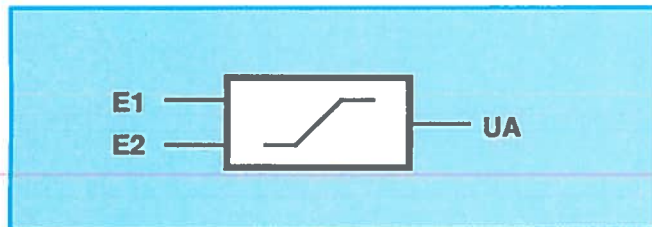


Fig. 71

Basically 2 functions can be realized with the limiting amplifier:

#### Limiting and Switching Amplifier

The individual functions can be subdivided as follows:

##### 1.) Limitation of analogue signals

This makes it possible (depending on the circuitry) to achieve unipolar or bipolar limiting functions.

2.) When the set values are exceeded, differential signals can be evaluated as a fault which depending on the circuitry can switch the amplifier positive or negative.

##### 3.) Switching amplifier for absolute signal detection

The inputs E1 and E2 have a summator function where the absolute value or the inverted absolute value (depending on circuit) is compared with the preset switching point.

4.) The signals can be stored or cleared by reset, as required.

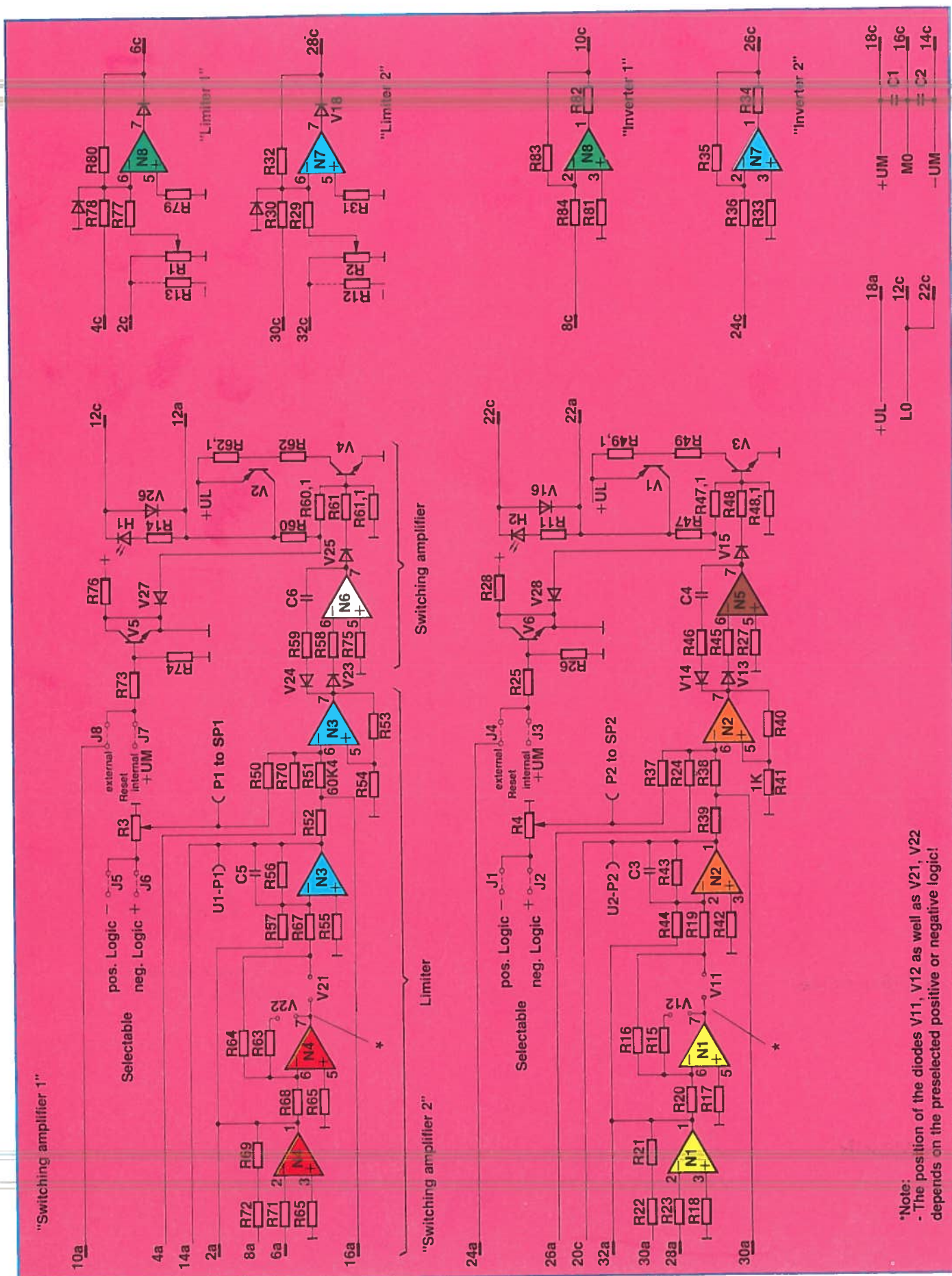


Fig. 72 Connection assignments of the limiter Type BG 1 S 30

## Determination of Measured Values

A particularly important point in a closed loop control circuit is the determination of the measured value, i.e. the actual value.

It is obvious that a system cannot be more accurate than the determination of the actual value. (Feedback). Measuring equipment should therefore be more accurate by a factor of 10 than the required accuracy of the system.

With regard to the obtainable degree of accuracy, the behaviour of the controlled system (dead time) must of course also be taken into consideration.

The measured value can be determined by digital means (figures) or analogue means (relative).

The definitions are explained with the aid of a position measurement example:

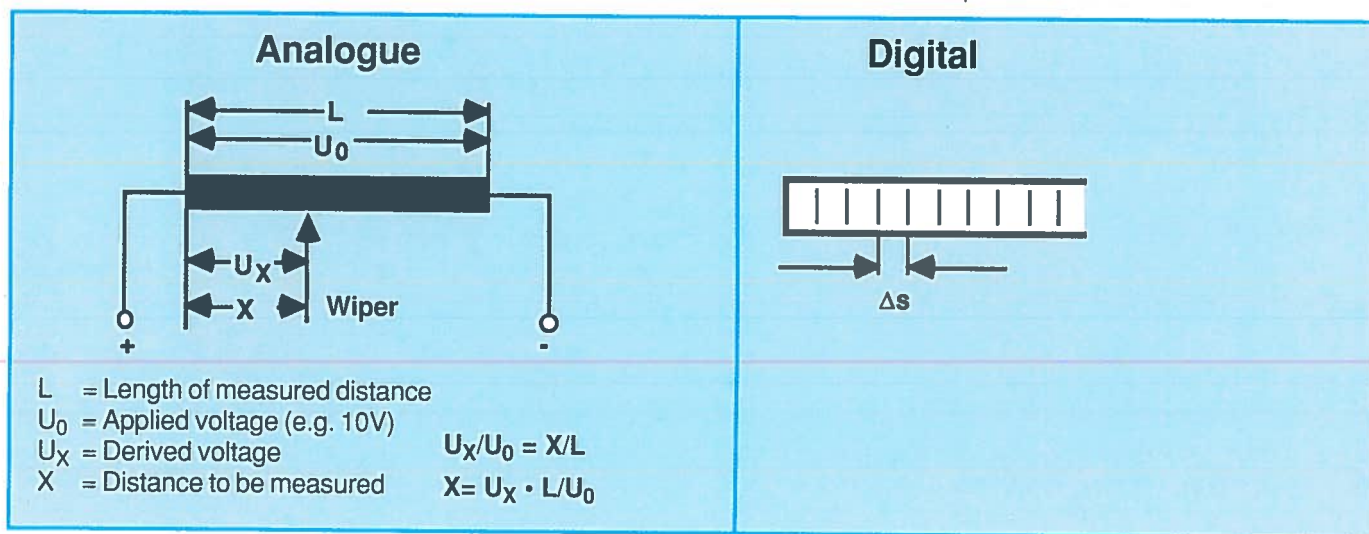


Fig. 73

**Digital:** Definition of measured values by means of unit steps defined by numbers

**Analogue:** Representation of a measured value in the form of a different analogue (relative) variable (e.g. voltage)

Further differentiation must be made between incremental and absolute measurement.

	Digital	Analogue
Incremental		(Analogue measurement is always absolute.)
Absolute	<p>Each line has a certain value.</p>	

Bild 74

**Incremental:** Increments are added or subtracted from a variable.

**Absolute:** Direct representation of a variable, in digital-absolute coded form (i.e. coded by symbols).

## 1 Position Measurement

### 1.1 Longitudinal Potentiometer Wire-wound

The distance is measured directly as an analogue variable in the form of voltage. The voltage signal is mostly between  $\pm 10 \text{ V} = 20 \text{ V}$ . The smallest usable signal is 20 mV. This, however, depends on the quality of the voltage supply, i.e. on the fluctuations which may occur so that in most cases 30 ... 50 mV can be considered as a useful signal.

Measured length: up to 500 mm

Example of measuring accuracy: 500 mm = 20 V

→ minimum measured length

$X = 500 \text{ (mm)} \cdot 0.02 \text{ (V)} / 20 \text{ (V)}$

$X = 0.5 \text{ mm}$

### 1.2 Conductive Plastic Potentiometer

This is a positional transducer with resistor and a collector line made of conductive plastic (analogue measurement).

Measured length: up to 1000 mm

Resolution: 0.01 mm

Also in this case, the degree of accuracy which can be obtained depends on the useful signal as described in 1.1. The advantages of this positional transducer are the low wear and the improved signal resolution (no winding jumps).

### 1.3 Inductive Positional Transducer (contactless)

In this measuring system, a moveable round bar made of magnetic soft steel is moved in a coil or in a coil system. The inductance of the measuring coil changes corresponding to the distance travel.

Measurement is carried out with alternating current in a bridge circuit. (Also see description of inductive positional transducers, Page D7).

Differential coils with an immersed core (Fig. 75) are suitable for measuring extremely small differences.

The sensitivity in this case is approx.  $2 \mu\text{m}$ .

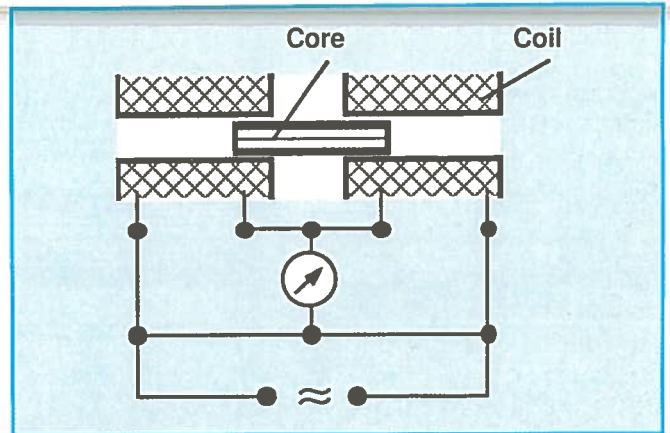


Fig. 75

### 1.4 Glass Scale

(NC length measuring system, photoelectric)

Measurement is digital-incremental in that grid type graduations on a scale are scanned photoelectrically (Fig. 76).

The photo elements generate periodical, almost sinusoidal signals as the scale moves relative to the scanning unit. The signals are evaluated in electronic circuitry.

Since, after the measuring system is switched off or in the case of power failure, the assignment of the measured value to the position is generally lost, the scale additionally features one or several reference marks. An additional signal (reference signal) is generated when passing over such a reference mark.

Measured length: 10 mm to 30 m (depending on system)

Accuracy:  $\pm 1 \mu\text{m}$  to  $\pm 10 \mu\text{m}$  (dependent on system)

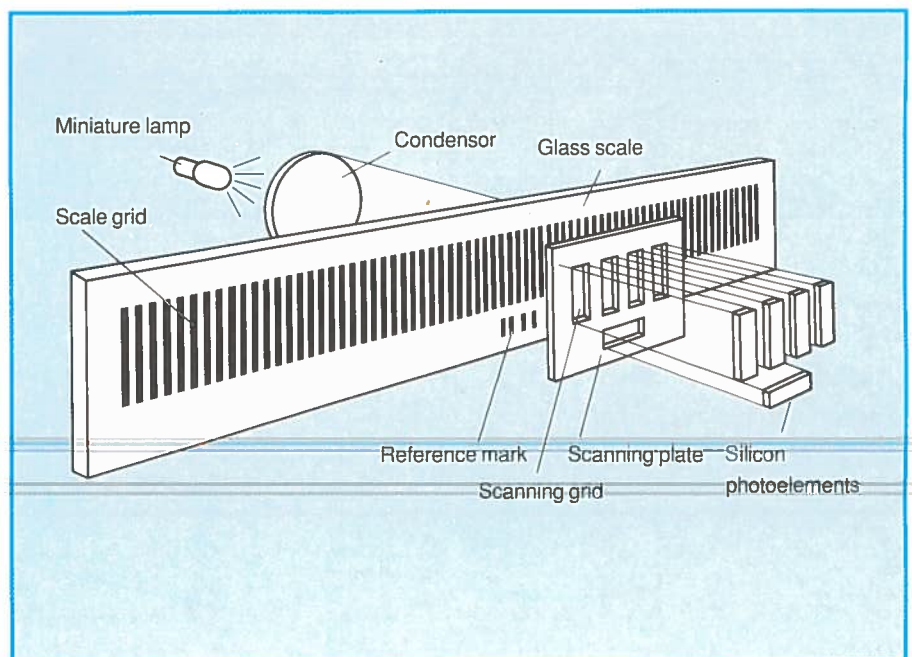


Fig. 76 Glass scale

### 1.5 Inductive Positional Transducer, Integrated in Hydraulic Cylinder

This positional measuring system is incorporated in the pressure chamber of a hydraulic cylinder.

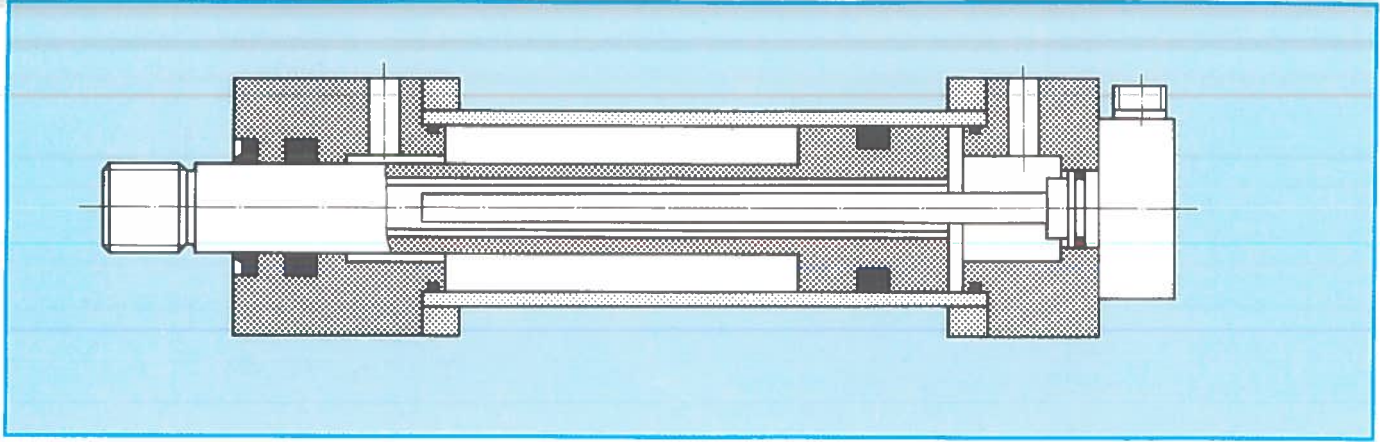


Fig. 77

Measured length up to 1000 mm are possible depending on the type of cylinder and piston diameter.  
Voltage supply: 2 to 5 V.

### 1.6 Ultrasonic Positional Transducer, Integrated in Hydraulic Cylinder

The measured, absolute value (distance) can be determined as often as necessary without it being corrupted by interruptions in operation, mains failure or other malfunctions.

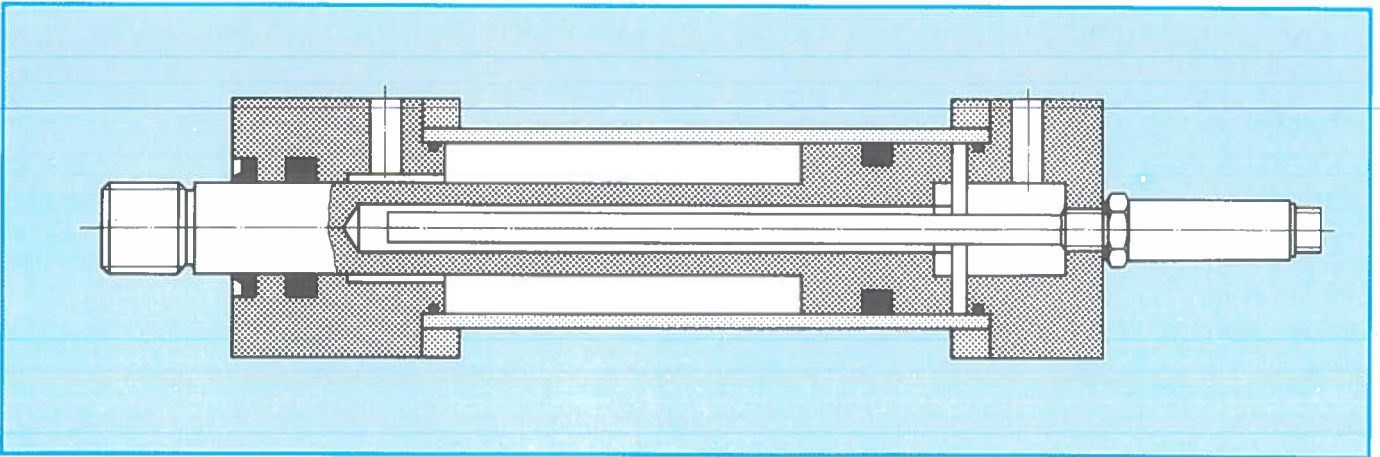


Fig. 78

The required positioning accuracy is in this case defined by the type of output signal:

- Analogue: 0 to 10 V DC
- Digital: Resolution 0.1 mm

Measured length up to 2500 mm.

### 1.7 Laser Measuring System

This measuring system is "proximity systems" used to determine workpiece dimensions or edge positions.

A transmitter generates a narrow band of laser light which is concentrated towards a detector in the receiver. Since this band of light consists of a fine, parallel beam, a workpiece placed in the measuring field throws a shadow related to a time base. The receiver determines the timed distances between the edges of the shadow and transfers these data to the microprocessor evaluator which in turn determines the workpiece dimensions.

Application examples of this measuring system:

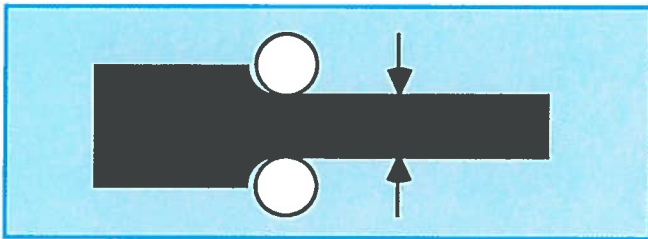


Fig. 79 Measuring the distance between rollers

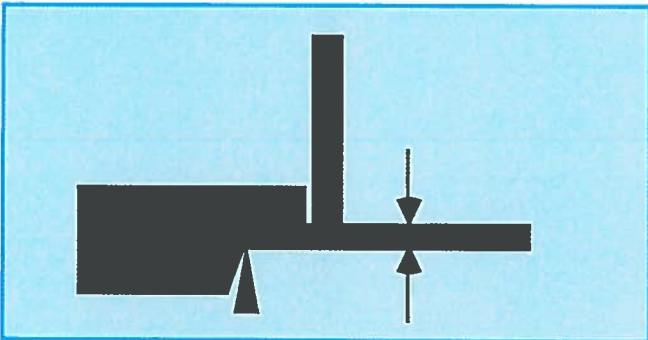


Fig. 80 Determining the edge position (tool position) referred to a reference edge

Measuring accuracy: as of  $\pm 0.25 \mu\text{m}$

Measures absolute size and deviation from nominal size.

### 1.8 Wire Strain Gauge

Wire strain gauges are sensors in which the length and cross section of a wire or film change as the strain load changes with the resistance also changing as a result.

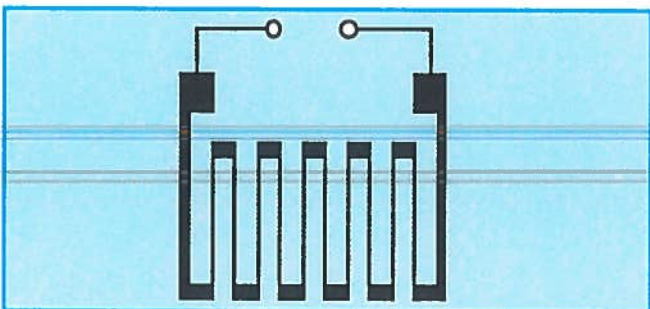


Fig. 81 Metal film measuring gauge

Normal wire strain gauge length are 3...60 mm. They can be used to measure changes in length of up to  $\pm 5$  o/o of their length.

### 1.9 Angle Encoder, Shaft Encoder

Position measurement is also possible with an angle encoder. The distance is represented as an angle by means of a rack and pinion, spindle and nut or measuring disc. Theoretically, the distance to be measured is unlimited.

### 2 Angle Measurement

#### 2.1 Rotation Potentiometer

The angle of rotation is represented as an analogue variable (voltage). The potentiometer can be wire-wound or a conductive plastic resistor.

The effective angle is up to approx.  $350^\circ$ , the maximum displacement velocity is up to 10000 rev/min.

The potentiometer receives a supply of  $\pm 10 \text{ V}$  (preferably from the output of an operational amplifier).

Minimum angle:  $\omega = 350^\circ \cdot 0.02 \text{ V} / 20 \text{ V} = 0.35^\circ$   
(with the smallest signal which can be evaluated of 20 mV).

#### 2.2 Incremental Angle Encoder

An incremental angle encoder generates a certain number of pulses per revolution. The number of pulses is a measure for the distance covered (angle or linear distance).

A code disc is mounted on a shaft. It is subdivided into individual segments which are alternately transparent and non transparent. The segments are scanned by infrared photo cells.

Since incremental angle encoders produce pulses continuously irrespective of the number of revolutions, large distances can be recorded without any problems.

Supply voltage: normally + 5 V DC

Minimum step:  $10 \mu\text{m}$

#### 2.3 Digital Absolute Angle Encoders (Digitizers)

Digitizers can be used in measuring and open loop control systems in which angular and linear displacement is to be measured. In such a system, the digitizer converts the rotary motion into electrical signals which are used for display or control purposes. Minimum step  $10 \mu\text{m}$ .

Basically measurements may be analogue or digital. The resolution of an analogue system is generally limited to  $10^{-3}$  or  $10^{-4}$  of the measuring range, whereas digital measuring systems achieve a much higher measuring accuracy. Furthermore, the result produced by the digital system is definitive and may easily facilitate further processing.

In this conjunction, a differentiation is made between two types of digitizers: Incremental and absolute. Incremental digitizers (pulse generators) generate periodic signals and to form a measured value they require

a memory (up-down counter) with the content of the memory defining the measuring range. Measuring errors, spurious external pulses and similar effects result in corruption of the measured value and cannot be corrected. When operation is interrupted or in the case of power failure, the memory is cleared and the measured value is lost.

In contrast to the incremental system, absolute digitizers are designed as coded measuring systems. In this case, a certain value can be absolutely assigned to each angular increment. Read out via a scanning element, this value can represent a numerical value. This means a measured value is not generated with the aid of auxiliary devices but rather it is represented unchanged as a code pattern. This absolute value is made available without losing time for further processing and as a measured value. It cannot be falsified as the result of interruption in operation or power failure. In this way, each position in a certain area (angle of rotation) is described with a code value which can be selected as often as required without the information content being falsified.

The functional principle is shown in *Fig. 82*. A rotating drive shaft carries a code disc which is positioned opposite a stationary aperture disc. The code disc has light-dark fields. The signals produced by the diode as a light transmitter are evaluated by the phototransistor as receiver. Depending on the version, the resolution of up to 4000 signals (data items) per revolution can be achieved.

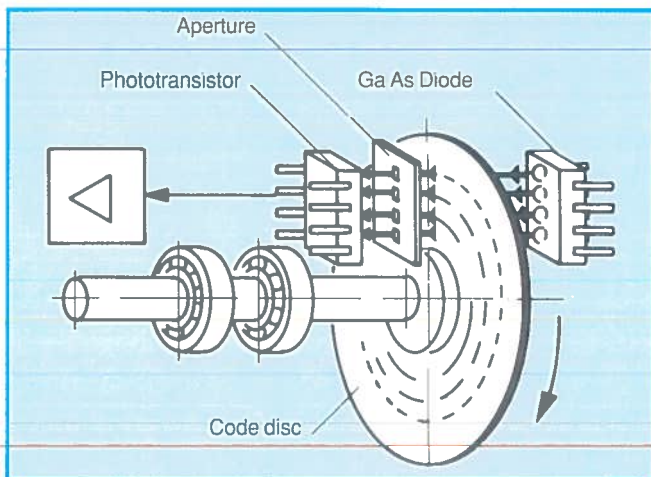


Fig. 82

### 2.4 Incremental Shaft Encoder

They are used to measure angle of rotation and angular velocity.

The version with glass scale corresponds to the system described under 1.4 for length measurement. Depending on the type of selected shaft encoder, a resolution of up to 100,000 measuring increments per revolution is possible.

## 3 Velocity Measurement

### 3.1 Tachogenerator

The tachogenerator produces a voltage dependent on speed which then acts as a measure for the rotary speed or can be converted in conjunction with a rack and pinion as a traverse speed.

Example:

Traverse speed  $v_{\max} = 1 \text{ m/sec}$ .

Ratio of the tachogenerator to the cylinder:

1 m cylinder stroke = 10 revolutions of the tachogenerator

(The ratio is selected such that the tachogenerator is operated within its normal range.)

e.g. 100 V at 1000 rev/min = 16.67 rev/sec

→ at  $v_{\max} = 1 \text{ m/sec}$ , the tachogenerator therefore produces an output voltage of

$$U = 10 \text{ (rev/sec)} / 16.67 \text{ (rev/sec)} \cdot 100 \text{ V} = \underline{60 \text{ V}}$$

### 3.2 Incremental Shaft Encoder

These are also used to measure angular velocity. As already described in connection with position measurement under Section 1.9 (also see 2.4 and 1.4), they are used in conjunction with rack/pinion, spindle/nut, lead screw or measuring wheel for velocity measurements. The increments are evaluated per unit of time.

### 3.3 Position Signal Differentiation

A further possibility of converting the velocity into a signal is differentiation of the position.

The analogue position signal is issued as a velocity signal via a differential element (D-element).

Accuracy approx. 2-3 %, referred to maximum voltage.

#### 4 Pressure Measurement, Force Measurement

##### 4.1 Pressure Transducer (strain gauge)

In this device, the pressure signal is transformed into an electrical signal by means of a form of strain gauge attached to a measuring element, i.e. a diaphragm.

The measuring range is from 0 to well over 1000 bars. The accuracy lies in the range  $\pm 0.2\%$  to  $\pm 0.5\%$ , dependent on, and referred to, full range value.

Basically, it is also possible to obtain indirect force measurement with any pressure measurement referred to an effective area, e.g. a cylinder.

Corresponding to the frequency range (depending on the type of pressure pick-up, e.g. up to 500 Hz or up to several thousand Hz) changes in pressure and therefore also pressure peaks can be measured in the ms range and below.

##### 4.2 Pressure Transducer with Inductive Positional Transducer

The movement of a diaphragm bending under load, can also be used to operate an inductive positional transducer to produce an electrical signal. Movement of the diaphragm is proportional to the effective pressure.

##### 4.3 Quartz Crystal Pressure Sensor Piezoelectric Load Cells

Pressure measurements with quartz crystals are particularly suitable for dynamic procedures, i.e. for determining pulsation and pressure peaks. Static pressure measurements, on the other hand, are possible only over a few minutes.

The operating principle incorporates the piezoelectric effect. If a force is applied to a quartz crystal in the direction of one of its three axes, then an electrical charge is produced at the surface positioned vertical with respect to the axis under load. This is proportional to the active load.

This voltage is now amplified and converted to a force or pressure value. Since the voltage follows, changes in force or pressure without any noticeable delay, as already mentioned, these transducers are particularly suitable for dynamic measurements.

The frequency range is between 10 to  $2 \cdot 10^5$  Hz.

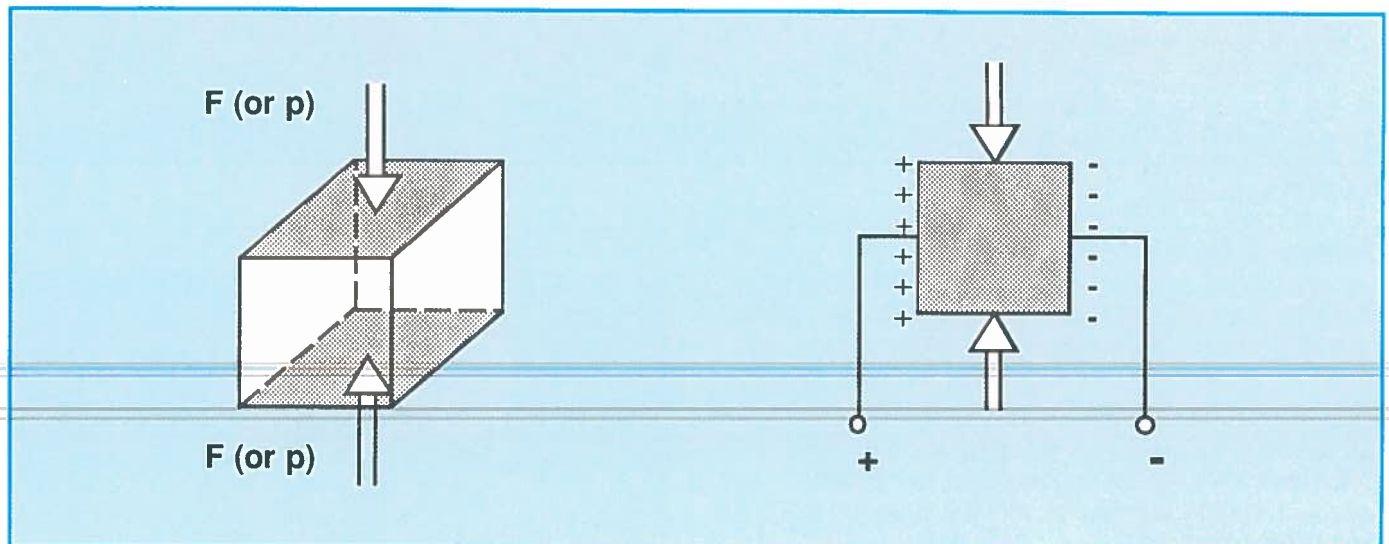


Fig. 83

### Appendix

The most important electronic modules which are often to be found in connection with open loop controls and particularly in closed loop control circuits are briefly described in the following.

#### Potentiometer

The potentiometer is an ohmic resistor with a variable tapping point (wiper). If, for example, a voltage of 0 V and 10 V is applied to the end of the potentiometer, then any value between 0 to 10 V can be tapped off at the wiper.

Example:

At a setting of 60 %, a voltage of 6 V can be tapped off at the wiper.

A potentiometer is used for

- setting a signal value  
i.e. the level of the tapped voltage corresponds to the required actual value as a distance, force or pressure.
- determining an actual value  
i.e. the tapped voltage value represents a distance and therefore a position.

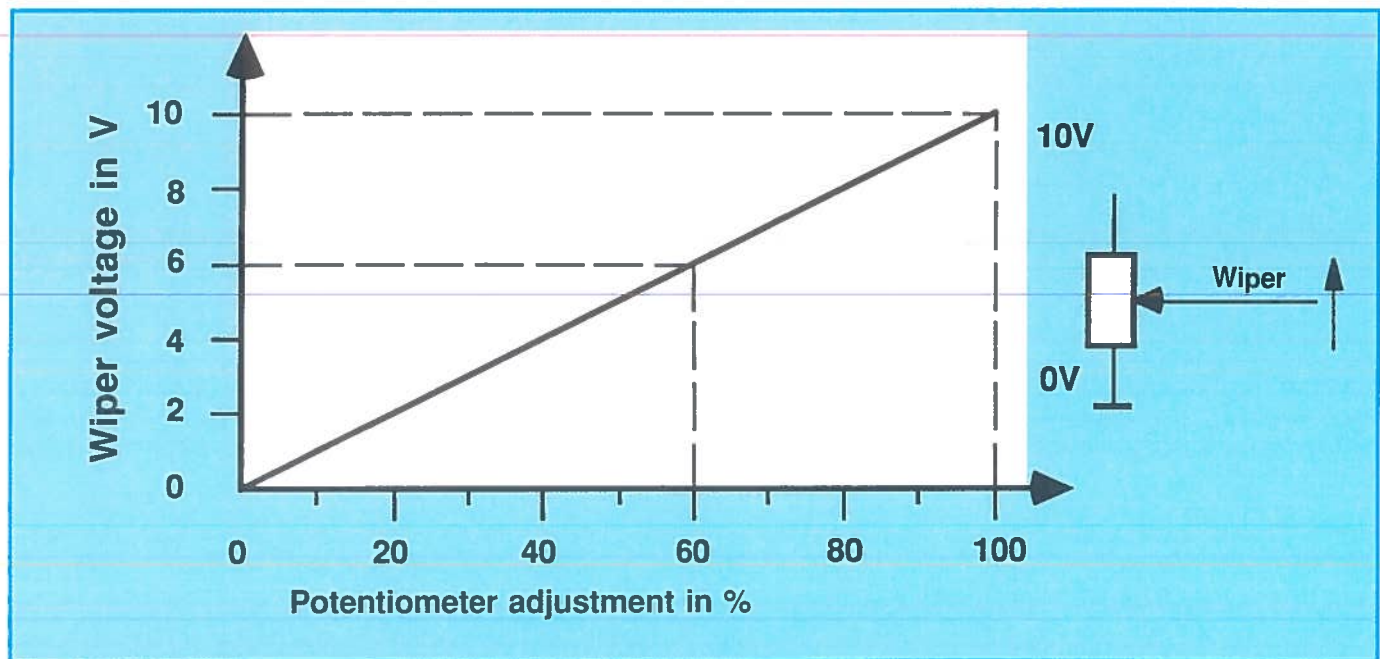


Fig. 84 Potentiometer

#### Operational Amplifier

The operational amplifier is a classic example of integrated circuit technology. (IC = integrated circuit). It is a multi-stage analogue amplifier with an extremely high amplification factor and is adapted to various tasks by means of external circuitry.

For example, by means of the suitable circuitry the following function units can be realized: Ramp generator, amplifier, inverter, summator, differentiator, limiter, the various controllers etc.

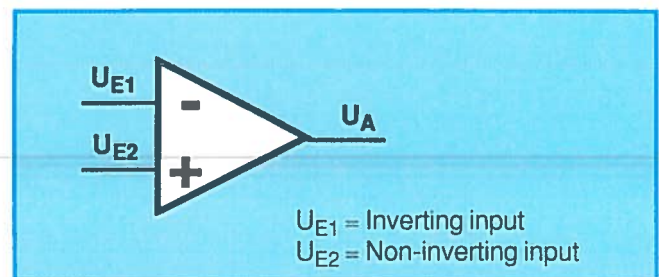


Fig. 85 Symbol for an operational amplifier

## Ramp, Ramp Generator

### Ramp Generator

The ramp generator produces a slowly rising or falling output signal from a stepped input signal. The change with respect to time of the output signal can be adjusted with a potentiometer. The functional principle of the ramp generator makes use of the fact that the capacitor  $C$  is charged slowly so that the output signal changes at a slow, steady rate, in response to a stepped input signal.

The rise of the output signal can be influenced with the variable resistor  $R$ , thereby determining the charging rate of the capacitor.

The set ramp time is always referred to 100 % signal value (stepped input signal).

Example:

Set ramp time is max. 5 sec at 100 % signal value. If, for example, a signal value of 60 % is set then the signal value is reached after approx. 3 sec.

In this way, the speed increase (acceleration) in a closed loop velocity control circuit or the speed in a closed loop position control circuit can be preset with a ramp generator. In a closed loop position control circuit, the set ramp time corresponds to the travel speed of the cylinder since the preset position is reached in this time.

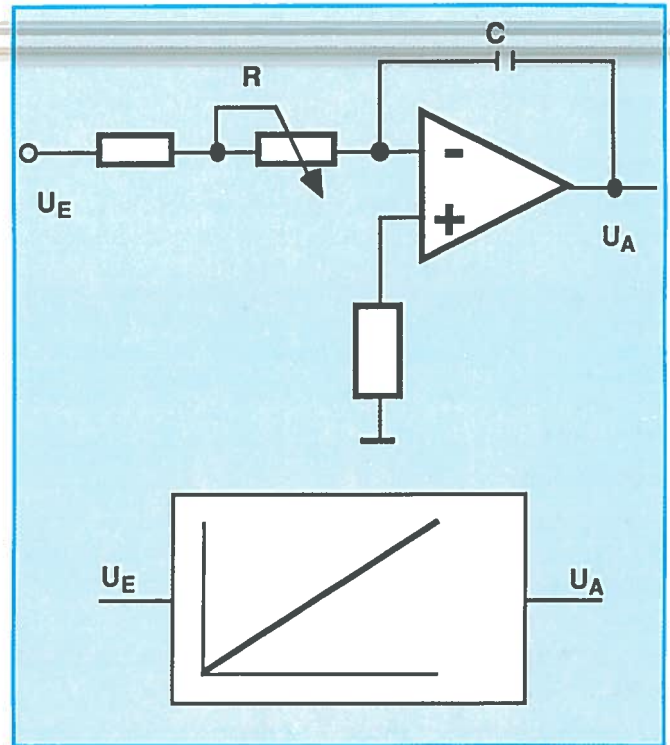


Fig. 86 Ramp generator

### Limiter

The applied input voltage is limited to a preset value as the output voltage. Limitation takes place via the two connections 1 (limitation of voltages smaller than zero) and 2 (limitation of voltages greater than zero).

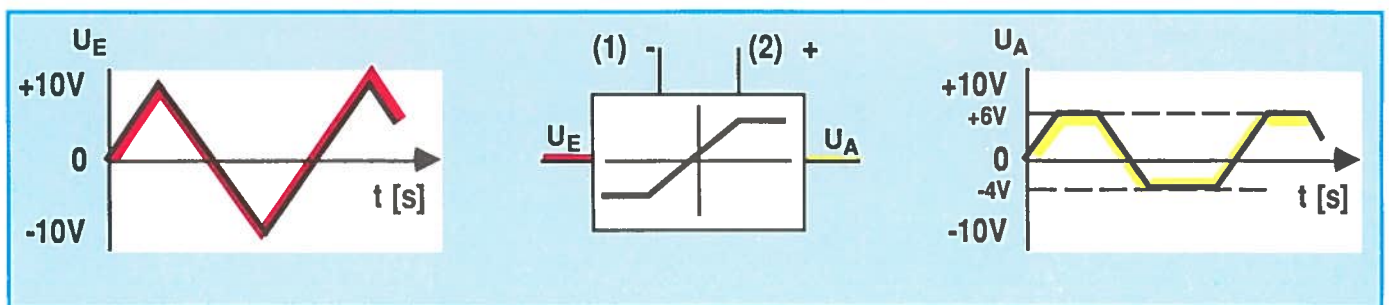


Fig. 87 Example of a limiter: Taking an input signal of  $U_E$  and an output signal of  $U_A$

### Controller

A controller refers to a device or component which carries out the essential processing of closed loop errors. The controller therefore compares the signal value with the actual value (feedback) and produces a corresponding output signal depending on the difference between the two values.

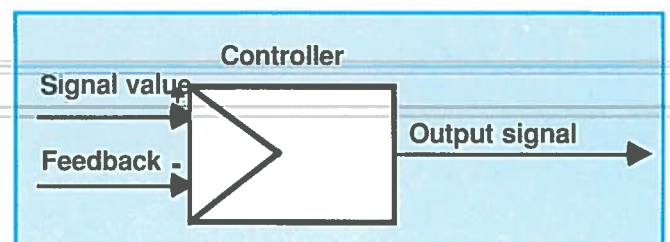


Fig. 88

## Amplifier

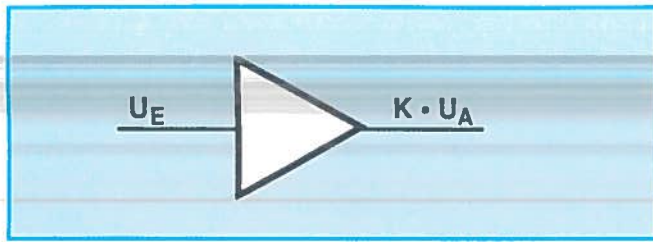


Fig. 89

The output voltage  $U_A$  is changed with respect to the input voltage  $U_E$  by the amplification factor  $K$ . Depending on the circuit, the polarity of the output voltage is usually reversed with respect to the input voltage.

## Inverter

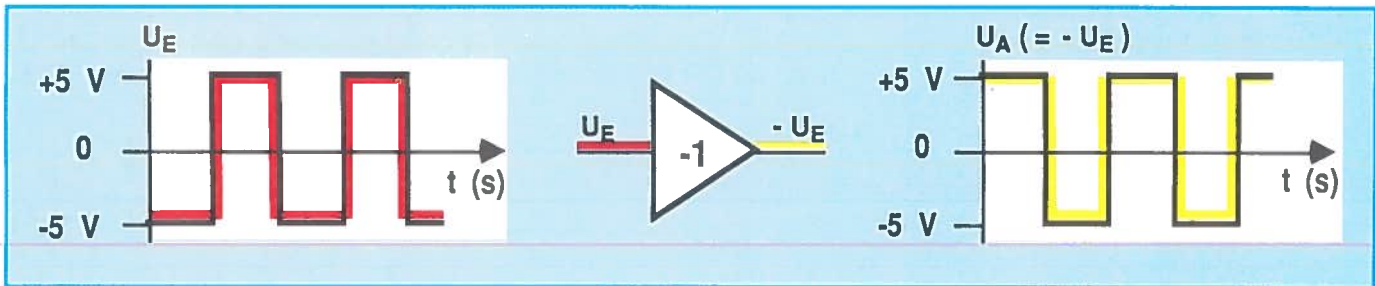


Fig. 90

The inverter reverses the polarity of the input voltage.

e.g.  $U_E = +5 \text{ V}$  - at output  $-U_E = -5 \text{ V}$

or  $U_E = -3 \text{ V}$  - at output  $-U_E = +3 \text{ V}$

It can therefore be considered as an amplifier with an amplification factor of  $-1$ .

## Power Amplifier

The input signal  $U_E$  is converted in the power amplifier into an output current which is proportional to  $U_E$ .

e.g.  $U_E$  0 to 10 V,  $I$  in mA = solenoid current.

## Matching Amplifier

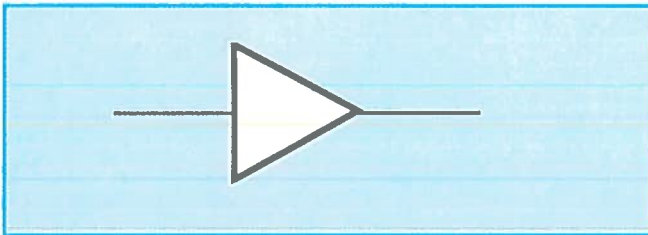


Fig. 91

With the aid of a matching amplifier, a voltage is produced by a measuring element (e.g. max. 60 V output voltage of a tachogenerator at maximum speed) is adapted to 10 V after the matching amplifier. This 10 V voltage then corresponds to a certain rotational speed or travel speed of a cylinder. Adaptation is necessary in order to further process the signal in a closed loop control circuit.

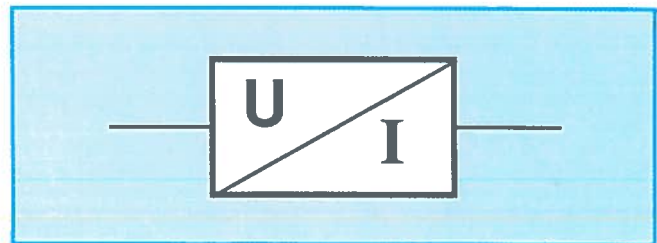


Fig. 92

### Schmitt-Trigger

Schmitt-triggers are used as threshold value switches. The two diagrams for input and output signal illustrate their function. If  $U_E$  exceeds a certain value ( $U_1$ ), then  $U_A$  jumps from one value to the other. Correspondingly, the output signal jumps back to the previous value (e.g. 0) as soon as  $U_E$  drops below a certain value ( $U_2$ ).

This results in two clearly defined switching points so that no switching takes place at intermediate values.

For example, if oscillations occur in the signal to a system, then these would be eliminated.

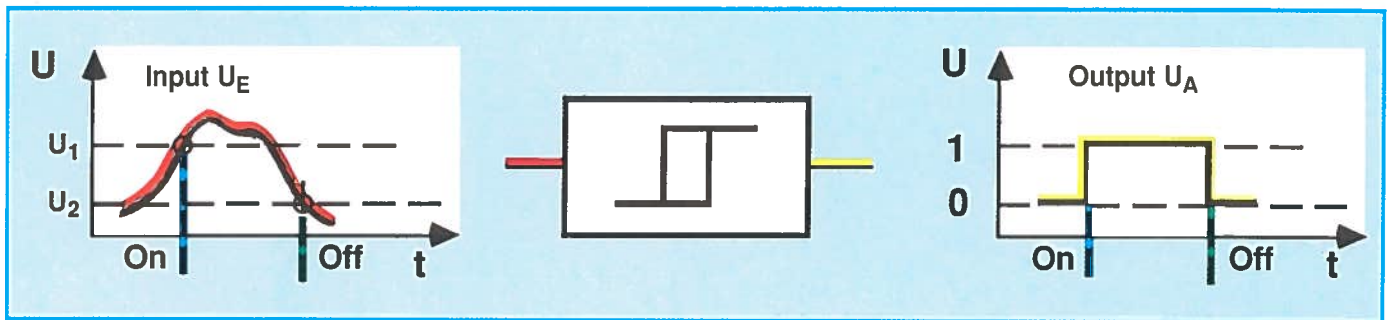


Fig. 93

### Summator (Adder)

With a summator, two signals can be added corresponding to their sign. It should be noted that the resulting output signal is inverted.

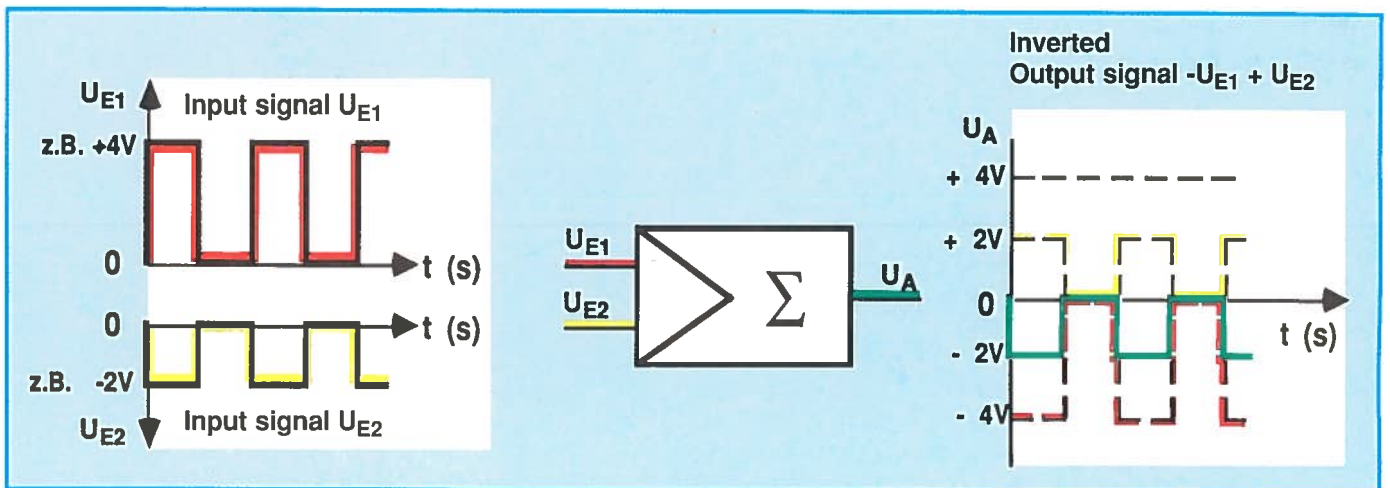


Fig. 94

Notes

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# **The Influence of the Dynamic Characteristics of the Servo Valve on the Closed Loop Control Circuit**

Dieter Kretz



## Preface

The following observations are intended to extend the understanding of the various relationships in the closed loop control circuit and to provide aids for accurately assessing the properties which can be expected of a control system.

Here, simple rules of thumb will be given instead of complicated mathematical considerations.

## Closed Loop Positional Control Circuit

Determining the effective loop gain " $K_{vopt}$ " and its influence on the closed loop control.

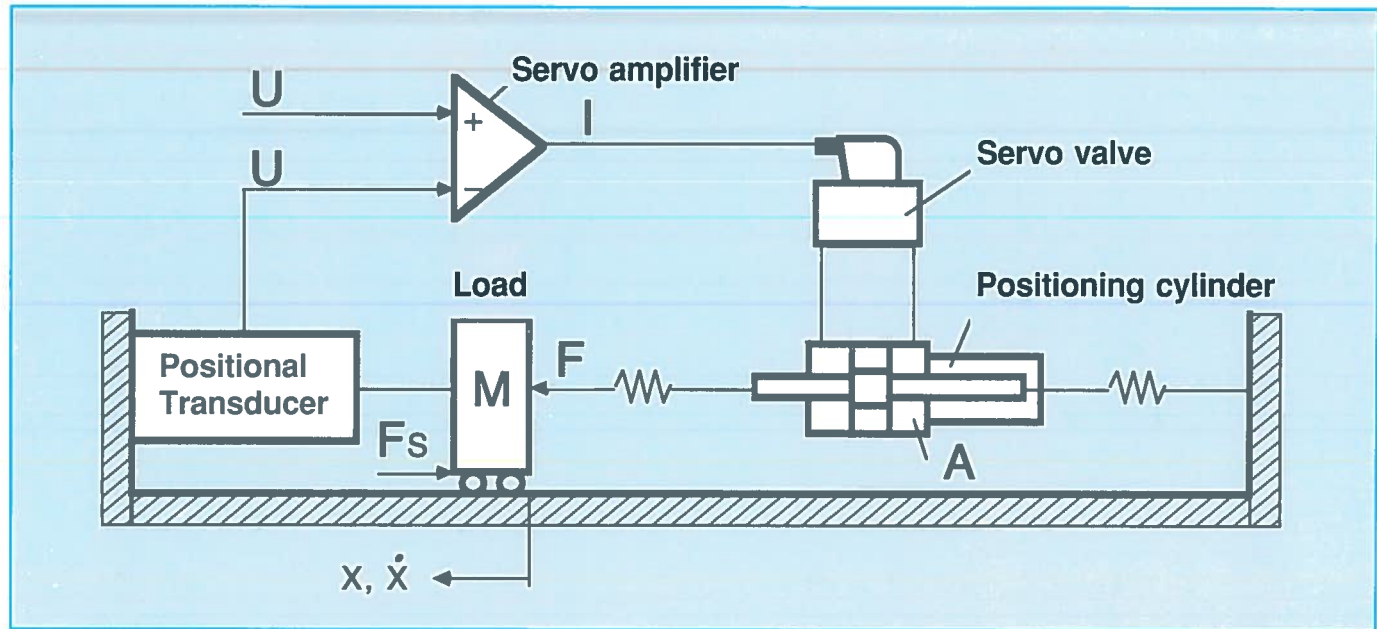


Fig. 1

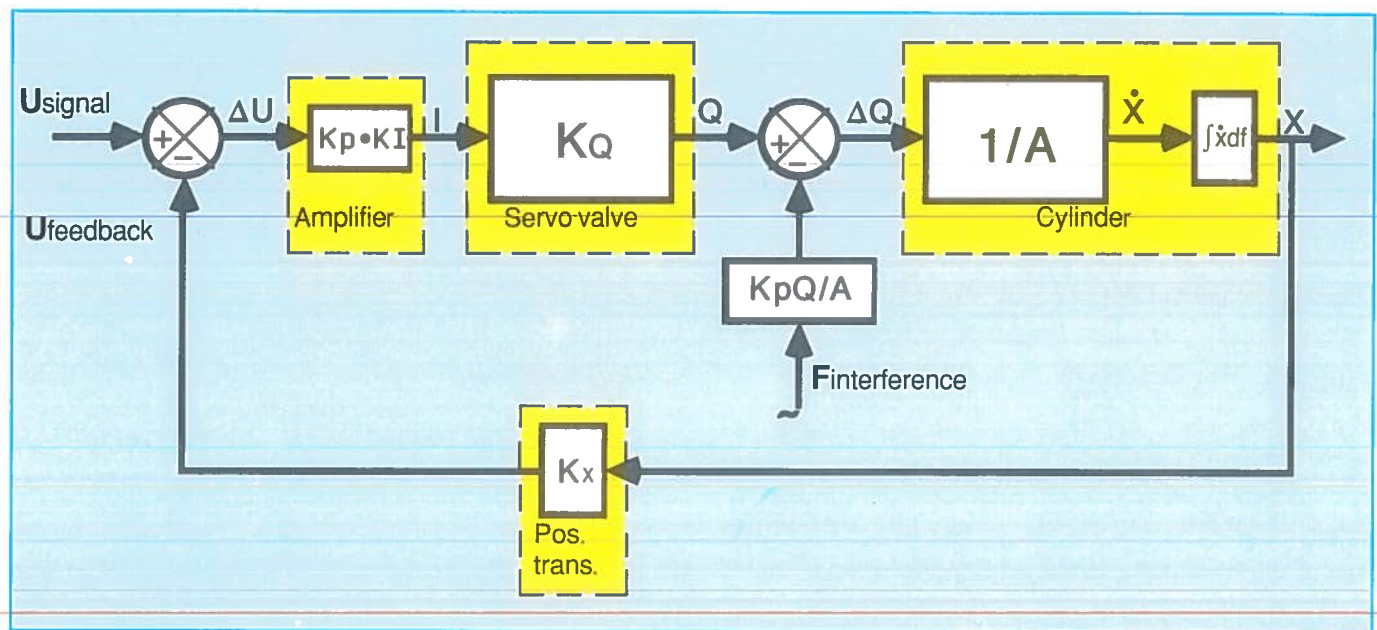


Fig. 2 Simplified block diagram

The loop gain  $K_v$  is equal to the product of the amplification factors of the transfer elements in the closed loop control circuits.

$$K_v = K_p \cdot K_i \cdot K_Q \cdot K_x / A \quad [s^{-1}]$$

- $K_Q$  = Flow amplification
- $K_P$  = Electrical amplification
- $K_i$  = Proportional amplification [mA/V]
- $K_x$  = Amplification of positional transducer [V/mm]
- $K_{PQ}$  = Pressure/flow gain (see Page H3)
- $A$  = Cylinder area

Fig. 3 shows a detailed representation of the frequency response equation in accordance with the Laplace transformation.

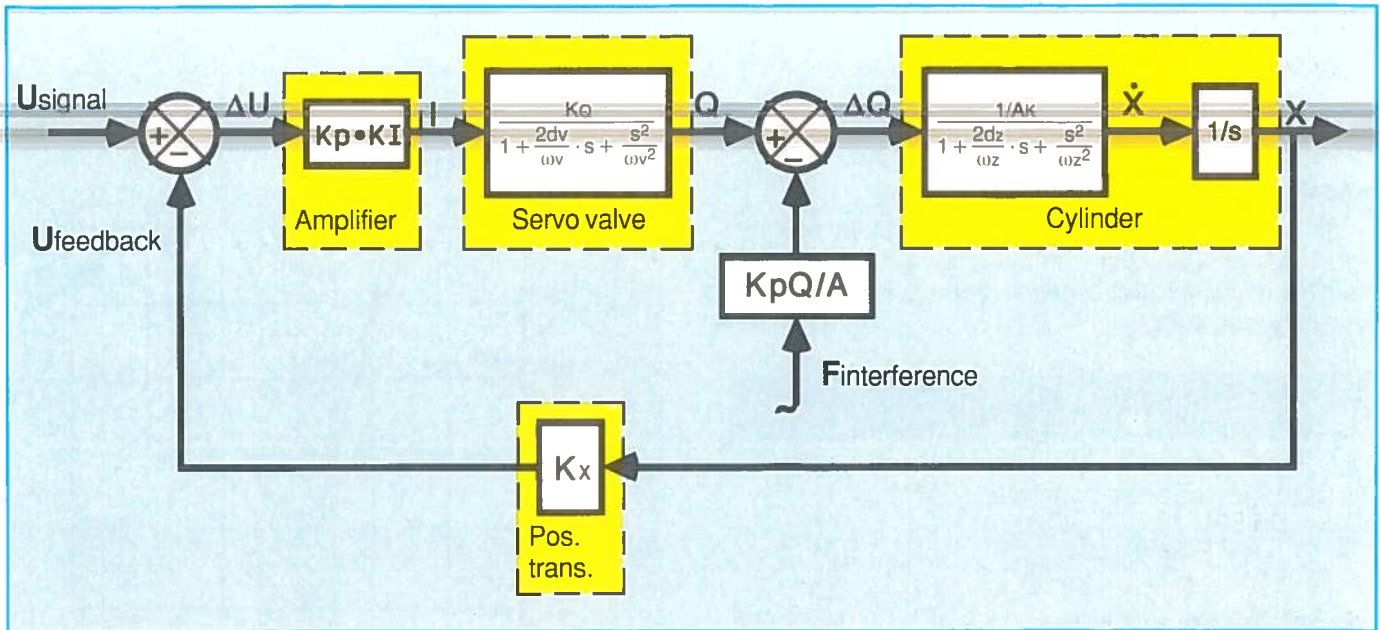


Fig. 3 Detailed block diagram

The valve and loaded cylinder are considered as 2nd order systems connected in series. The characterizing feature of the servo drive (cylinder) is the nominal hydraulic frequency (mass/hydraulic spring). The transition from velocity to positioning distance is represented by the integration (1/s).

#### Time Constant of Closed Loop Control Circuit

The time constant is proportional to  $1/K_v$

$$T = 1/K_v \quad [\text{s}]$$

$K_v$  = Loop gain

That is, the greater the loop gain  $K_v$  the faster the system.

#### Stiffness

When stationary, the stiffness with respect to force interference is expressed as

$$C = F_{\text{disturbance}}/X = K_v \cdot A^2/K_{pQ}$$

The stiffness is therefore proportional to the loop gain and inversely proportional to the flow amplification  $K_{pQ}$ .

$$K_{pQ} = V_Q/V_p \quad [\text{cm}^3/\text{s}/\text{bar}]$$

$V_Q$  = Flow gain ( $\text{cm}^3/\text{s}$ )  
 $V_p$  = Pressure gain (bar)

This represents the flow/pressure amplification of the valve plus a pressure-dependent leakage at the driven unit. An increase in the area of the positioning piston or of the capacity of the servo motor increases the stiffness proportional to the square of the increase.

#### Positioning Error

Normally, less than 5 % of valve current is required in a closed loop control to correct a positional error due to a disturbing factor, or cause the last positioning movement to bring the velocity to zero. The reason is, that at the very latest, the full system pressure is available at 5% signal (opening) to perform the correction (see pressure amplification, page F7).

The positioning error is therefore

$$\Delta X \leq 0.05 \cdot v_{\text{max}}/K_v \quad [\text{mm}]$$

It can be seen that the loop gain should be as large as possible.

The greater  $K_v$  is selected the smaller the positioning error and therefore the stiffer the system towards interfering forces.

$v_{\text{max}}$  represents the velocity at 100 % opening of the servo valve.

Consequently, it can be seen that the nominal flow of the servo valve  $Q = A \cdot v_{\text{max}}$  should be selected as small as possible.

For stability reasons, the loop gain cannot be selected at will.

If the loop gain  $K_v$  is greater than a critical circuit frequency  $K_{v \text{ crit}}$  the system will oscillate, i.e. the system will be unstable.

## What is the Maximum Value of $K_V$ ?

A differentiation can be made between 2 cases:

a) The servo valve frequency  $\omega_V$  (frequency at  $-90^\circ$  phase offset) is considerably higher than the natural frequency of the load  $\omega_L$ .

In this case, the dynamics of the part system with the higher natural frequency can be neglected. As a result, the closed loop control circuit is reduced to a 3rd order system expressed as:

$$K_V < K_{Vcrit} = 2 D \omega_L$$

That is  $K_V$  must be selected smaller than  $K_{Vcrit}$ .  
 $D$  = dimensionless damping factor.

Fig. 4 shows the time ratio of such a 3rd order closed loop control where the relative attenuation and relative gain serve as parameters. The optimum value  $K_{Vopt}$  is normally derived from this time ratio, i.e. from the stepped response. If  $K_V$  is kept small at the given attenuation, a rather uniformly increasing stepped response results; if, on the other hand,  $K_V$  is extremely large, intense superimposed oscillation occurs.

The quality criteria can be defined based on the progression of this stepped response (transient response). The ITAE criterion (Integral of Time multiplied by Absolute Error) is often used:

$$ITAE = \int_0^{\infty} t |X_E - X_A| \cdot dt$$

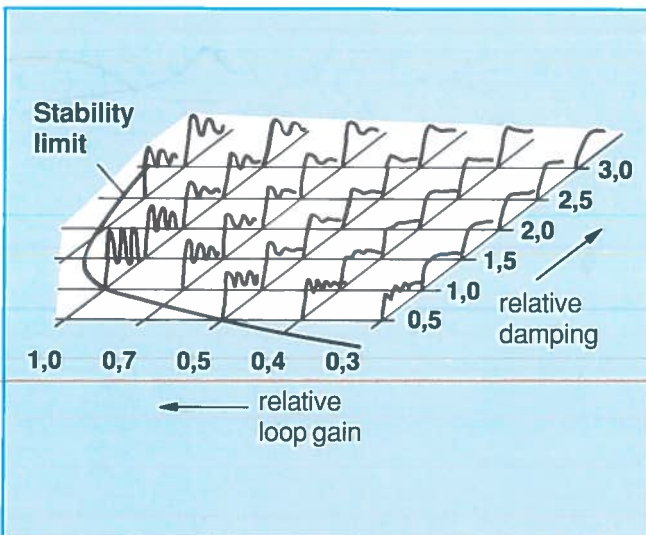


Fig. 4 Time ratio of a 3rd order closed loop control

The loop gain at which this ITAE value is minimum is then considered as the optimum. If  $K_V$  is varied at constant level of damping and if the ITAE value is distributed over the relative gain  $K_V/\omega_L$ , Fig. 5 results.

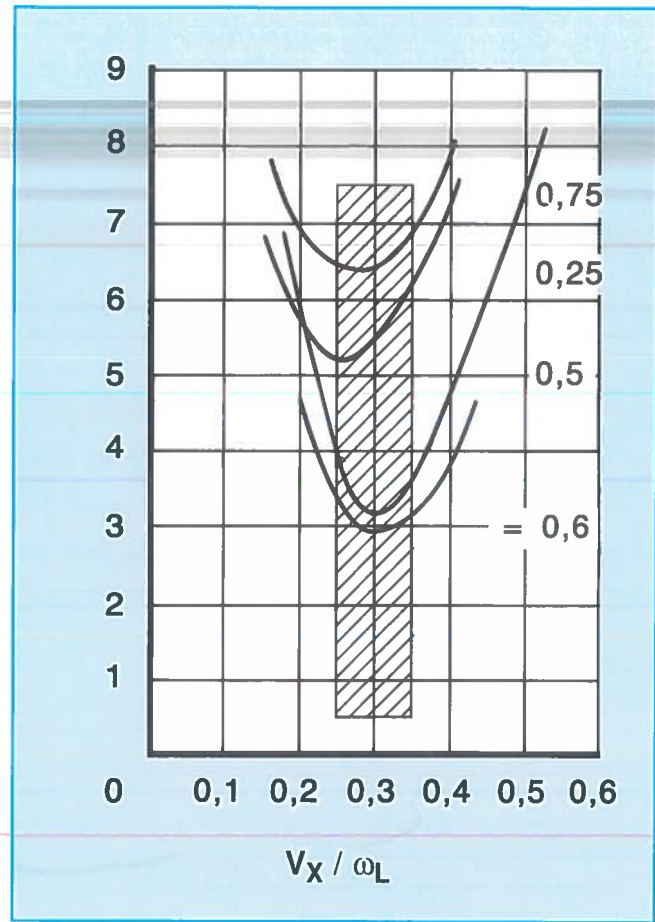


Fig. 5

It can be seen that for the range of typical damping coefficients of attenuation ( $0.2 < D < 0.9$ ), the optimum ITAE values are between  $K_V/\omega_L = 0.25$  and  $0.35$ .

This results in Rule 1

$$K_{Vopt} = 1/3 \cdot \omega_L \quad [s^{-1}]$$

This gain, also termed velocity gain, is the product of the hydraulic amplification and electrical amplification.

## b) Both Natural Frequencies are taken into Consideration

This results in a 5th order system. Stability considerations produce a critical frequency  $\omega_{crit}$  and a critical loop gain  $K_{Vcrit}$  which are dependent on the two natural frequencies  $\omega_V$  = natural valve frequency and  $\omega_L$  = natural load frequency.

The critical frequency  $\omega_{crit}$  is always smaller than the smaller of the two frequencies  $\omega_V$  and  $\omega_L$ .

Neglecting the attenuation factors Rule 2 is

$$\omega_{crit} = \omega_V \cdot \omega_L / (\omega_V + \omega_L)$$

The optimum loop gain is therefore

#### Rule 3

$$K_v \text{ opt} = 1/3 \omega_{\text{crit}}$$

Accuracy of position and stiffness with respect to interfering forces require a high electrical amplification  $K_p$ .

On the other hand, the hydraulic amplification should be selected only as large as necessary (see positioning error).

#### Rule 4

Use valve with the smallest possible nominal flow. Normally, this is also the valve with the higher dynamics.

### Determining the Flow Frequencies

#### Servo Valve

The frequency response of the servo valve is obtained from the frequency response curve.

#### Cylinder

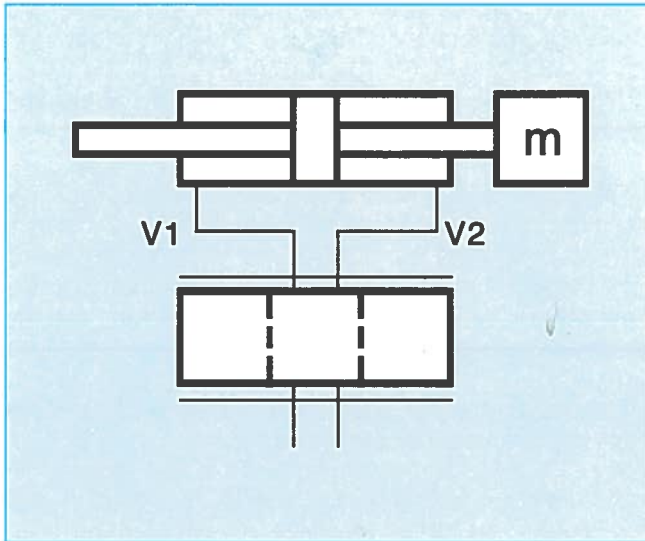


Fig. 6 Determining the natural frequency with double rod cylinder

$E$	= Modulus of elasticity of oil	$[\text{kg/cm} \cdot \text{s}^2]$
	$1.4 \cdot 10^7$	
$A_R$	= Annulus area of cylinder	$[\text{cm}^2]$
$H$	= Cylinder stroke	$[\text{cm}]$
$V$	= Total oil volume	$[\text{cm}^3]$
$m$	= Mass	$[\text{kg}]$
$V_{LR}$	= Oil volume in the pipe on the annulus side of the cylinder	$[\text{cm}^3]$
$\omega_0$	$= \sqrt{2 \cdot E \cdot A_R^2 / (V \cdot m)}$	$[\text{s}^{-1}]$
$V$	$= V_1 = V_2 = A_R \cdot H/2 + V_{LR}$	$[\text{cm}^3]$

The natural frequency is minimum in the centre position of the cylinder.

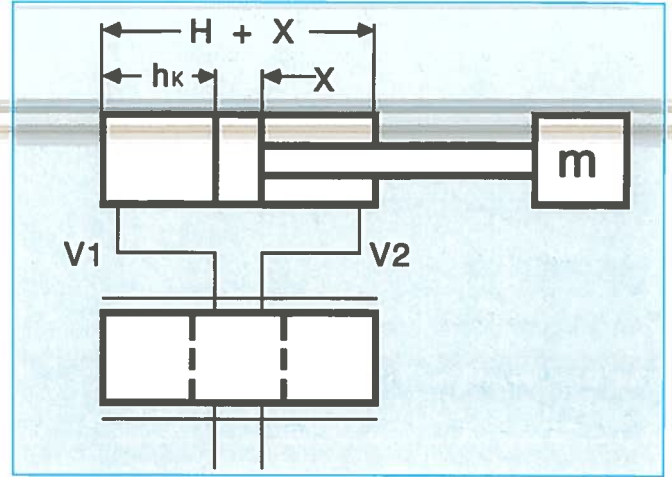


Fig. 7 Determining the natural frequency with single rod cylinder

$E$	= Modulus of elasticity of oil	$[\text{kg/cm} \cdot \text{s}^2]$
	$1.4 \cdot 10^7$	
$A_R$	= Annulus area of cylinder	$[\text{cm}^2]$
$A_K$	= Piston area of cylinder	$[\text{cm}^2]$
$V_1$	= Oil volume on piston side	$[\text{cm}^3]$
$V_2$	= Oil volume on annulus side	$[\text{cm}^3]$
$m$	= Mass	$[\text{kg}]$
$H$	= Stroke	$[\text{cm}]$
$h_K$	= Cylinder stroke at min. natural frequency	$[\text{cm}]$
$V_{LK}$	= Pipe volume on piston side	$[\text{cm}^3]$
$V_{LR}$	= Pipe volume on annulus side	$[\text{cm}^3]$
$\omega_0$	$= \sqrt{(C_1 + C_2)/m}$	
$\omega_0$	$= \sqrt{E \cdot A_K^2 / (V_1 \cdot m) + E \cdot A_R^2 / (V_2 \cdot m)}$	$[\text{cm}^3]$
$V_1$	$= A_K \cdot h_K + V_{LK}$	$[\text{cm}^3]$
$V_2$	$= A_R \cdot (H - h_K) + V_{LR}$	$[\text{cm}^3]$

$$h_K = \frac{\left( \frac{A_R \cdot H/10}{\sqrt{A_R^3}} + \frac{V_{LR}}{\sqrt{A_R^3}} - \frac{V_{LK}}{\sqrt{A_K^3}} \right)}{\left( \frac{1}{\sqrt{A_R}} + \frac{1}{\sqrt{A_K}} \right)} \cdot 10 \quad (\text{mm})$$

The natural frequency is minimum in cylinder position  $h_K$ .

#### Hydraulic Motor Natural Frequency

$\omega_0$	$= \sqrt{2 \cdot (q/2 \cdot \pi)^2 \cdot E / (V_1 \cdot J)}$	
$q$	= Capacity	$[\text{cm}^3]$
$V_1$	= Oil volume	$[\text{cm}^3]$
$J$	= Moment of inertia	$[\text{kgcm}^2]$
$E$	= Modulus of elasticity of oil	$[\text{kg/cm} \cdot \text{s}^2]$
	$1.4 \cdot 10^7$	

If the calculation of the drive shows that the accuracy requirements are not met, the loop gain can be increased by corresponding control circuitry.

The following circuits make it possible to increase the optimum loop gain and therefore to improve the positioning accuracy.

- Controller circuit as PD-controller
- Feedback of load pressure
- Feedback of velocity
- An integral circuit can increase the accuracy to any degree, at the same time, however, requirements regarding dynamics limit the I-component.
- An increase in gain also enables an increase in damping by means of a bypass leakage between the load connections. The static stiffness is, however, reduced as a result.

### Selection of Measuring System

As already mentioned, a measuring system is necessary in order to control a physical variable. It must be possible for the system to convert the relevant variable into an electrical signal - current or voltage. Correspondingly, devices are required to measure differences, angles, velocity, speeds, pressure, forces, torque and acceleration. A number of measuring principles can be implemented for each of these variables. They are used taking into consideration the measuring range, accuracy requirements, service life, ambient conditions etc. The number of measuring elements is correspondingly large, making it impossible to provide even a general overview.

The following is however generally applicable:

- A closed loop control can never be more accurate than the measuring method.
- The measuring system is characterized by its transmission factor. This is the relationship of the output voltage or current to the measured variable.
- The accuracy of the measuring system must be at least 5 times greater than the required accuracy of the closed loop control.
- The measuring system must be able to follow the changing variable without delay.
- The transmission factor and the neutral point must remain constant under all operating conditions.
- The electrical signal must be processed such that it is free or can be kept free of interferences caused by neighbouring high power elements.
- The coupling of the measuring system with the drive must be extremely stiff and free of play.
- The measuring system must be arranged such that the control variable is registered directly and not falsified by secondary effects.

These few points clearly illustrate the significance the measuring system has for closed loop control technology and for servo hydraulics.

## Calculation Example

50/36 x 100 stroke

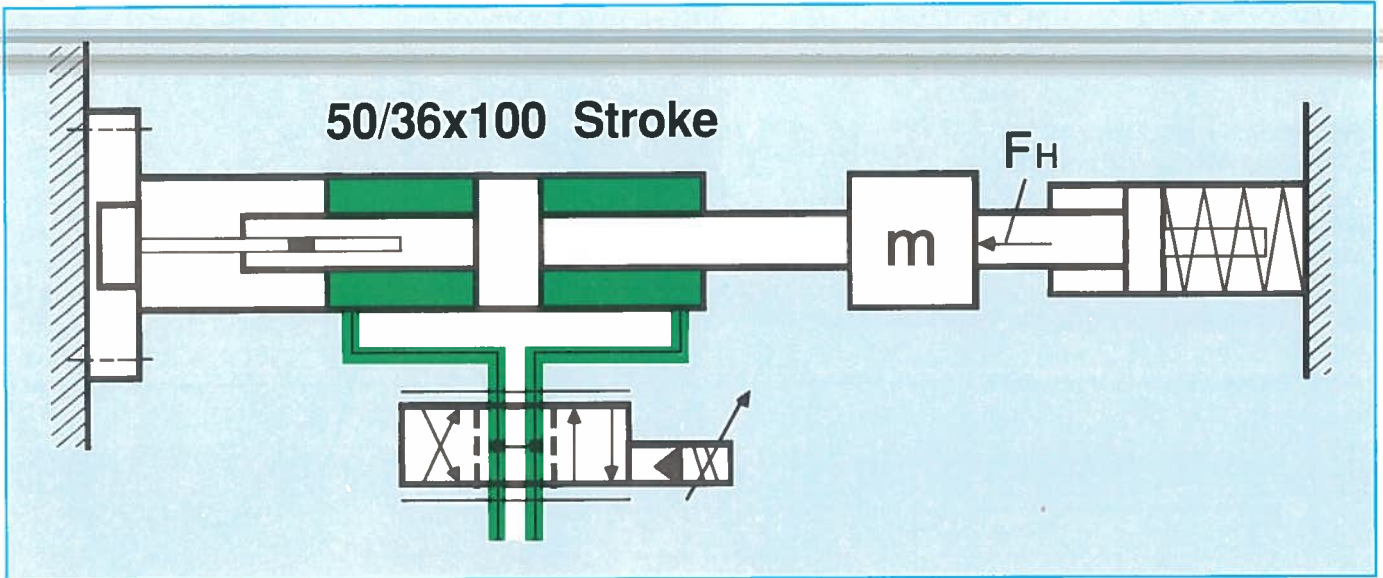


Fig. 8

Cylinder 50/36 x 100 stroke

Cylinder annulus area

$$A_R = 9.45 \text{ cm}^2$$

Mass moved

$$m = 500 \text{ kg}$$

Traverse time for 80 mm stroke

$$t = 400 \text{ ms}$$

Stopping force

$$F_H = 6000 \text{ N}$$

### Hydraulic Natural Frequency of Cylinder/Mass System

$$\omega_0 = \sqrt{2 \cdot E \cdot A_R^2 / (V \cdot m)}$$

If the valve is mounted directly on the cylinder, the volume will be

$$V = H/2 \cdot A_R$$

used in the above formula for  $V$  the result is

$$\omega_0 = \sqrt{4 \cdot E \cdot A_R / (H \cdot m)}$$

$$\omega_0 = \sqrt{4 \cdot 1.4 \cdot 10^7 \text{ (kg/cm s}^2\text{)} \cdot 9.45 \text{ (cm}^2\text{)} / (10 \text{ (cm)} \cdot 500 \text{ (kg)})}$$

$$\omega_L = \omega_0 = 325 \text{ s}^{-1}$$

$$f_0 = 51 \text{ Hz}$$

For the case that the natural valve frequency is considerably higher than the natural frequency of the cylinder/mass system, the loop gain  $K_v$  will be

$$K_v < K_{v \text{ crit}} = 2 D \cdot \omega_L \text{ (see Page J4 Case a)}$$

### Rule 1

$$V_{\text{opt}} = 1/3 \omega_L$$

$$V_{\text{opt}} = 325/3 = 108 \text{ s}^{-1}$$

### Time Constant

$$T = 1/V = 1/108 \text{ s}^{-1} \approx 0.0092 \text{ s}$$

### Possible Acceleration Time

$$T_B = 5 \cdot T \approx 50 \text{ ms}$$

### Selection of Servo Valve

#### Maximum Velocity

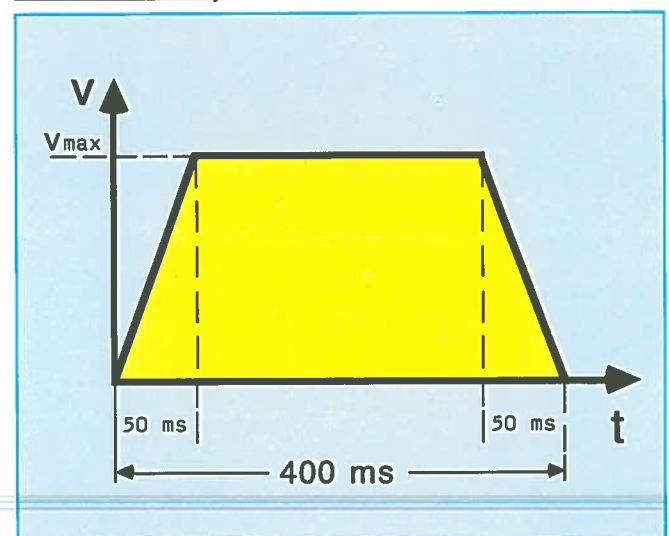


Fig. 9

$$v_{\text{max}} = s / (T_{\text{ges}} - T_B) = 80 \text{ (mm)} / (0.4 \text{ (s)} - 0.050 \text{ (s)})$$

$$v_{\text{max}} = 228 \text{ (mm/s)}$$

### Required Flow

$$Q = A \cdot v = 9.45 \text{ (cm}^2\text{)} \cdot 22.8 \text{ (cm/s)}$$

$$Q = 215.5 \text{ (cm}^3\text{/s)}$$

$$Q = 13 \text{ (l/min)}$$

Selected:

Servo valve with  $Q_N = 20 \text{ l/min}$  at  $\Delta p = 70 \text{ bar}$ .

### Calculation of Loop Gain Taking into Consideration the Natural Valve Frequency

Natural valve frequency from the frequency response  
(see Fig. 10)

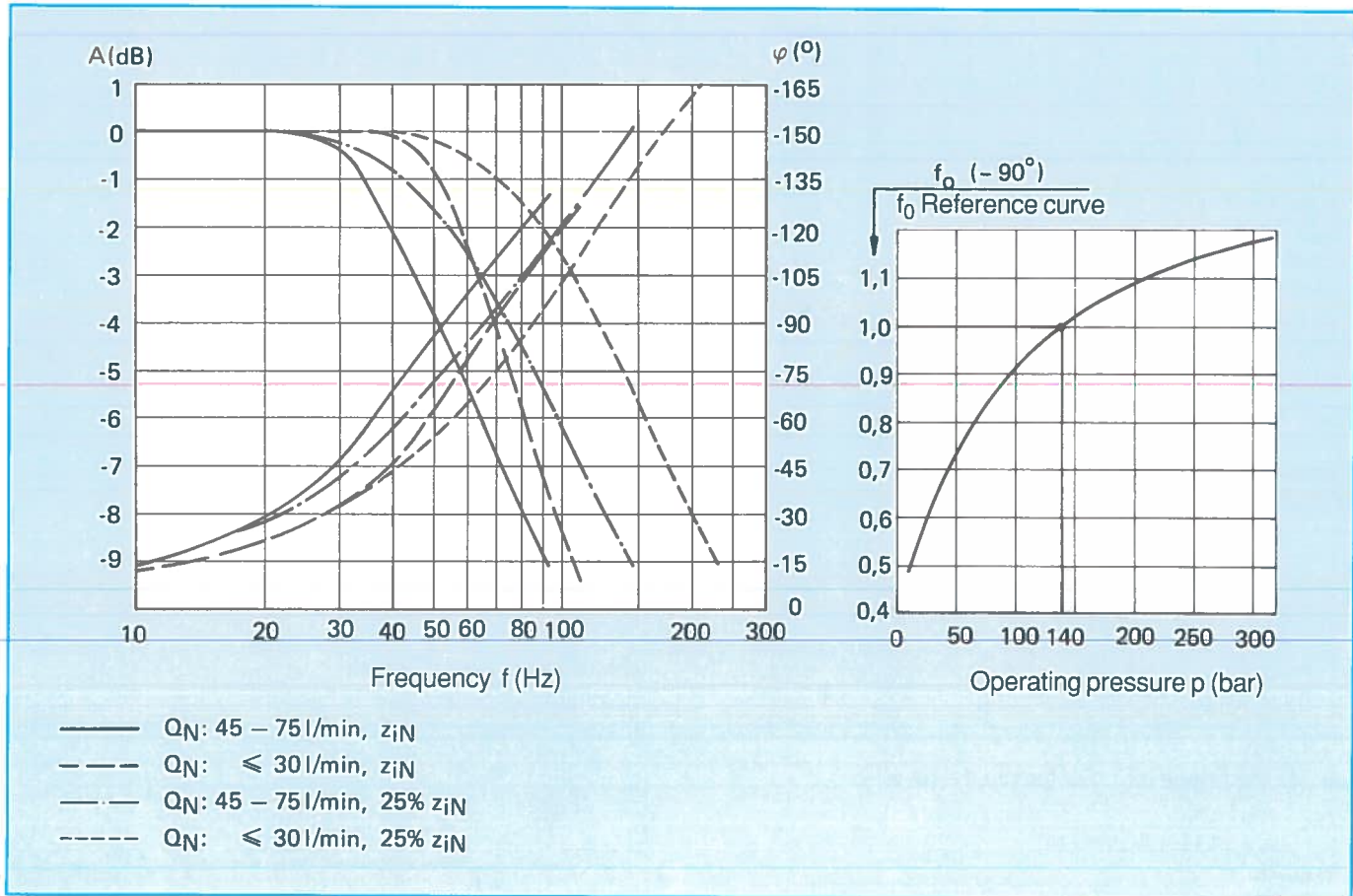


Fig. 10 Typical frequency response reference curves (left) and frequency dependency of operating pressure (right) for servo valves with mechanical feedback

#### Rule 2

$$\omega_{\text{crit}} = \omega_V \cdot \omega_L / (\omega_V + \omega_L)$$

$$\omega_{\text{crit}} = 534 \cdot 325 / (534 + 325)$$

$$\omega_{\text{crit}} = 202 \text{ [1/s]}$$

#### Determining $\omega_V$ from the Frequency Response

For  $\leq 30 \text{ l/min}$  and  $25\%$  signal  
 $f_{90^{\circ}} = 85 \text{ Hz}$  at  $140 \text{ bar}$

$$\omega_L = 2 \cdot \pi \cdot 85 = 534 \quad [1/s]$$

#### Rule 3

$$K_{\text{vopt}} = 1/3 \omega_{\text{crit}} = 202/3 = 67.3 \text{ [1/s]}$$

Comparison of the two calculated loop gain factors shows that, in this case, the valve greatly influences the possible loop gain and must therefore be taken into consideration.

#### Time Constant

$$T = 1/K_v = 1/67 \text{ (1/s}^{-1}\text{)} = 0.015 \text{ [s]}$$

#### Possible Acceleration Time

$$T_B = 5 \cdot T = 0.075 \text{ [s]}$$

#### Selection of Servo Valve

##### Maximum Velocity

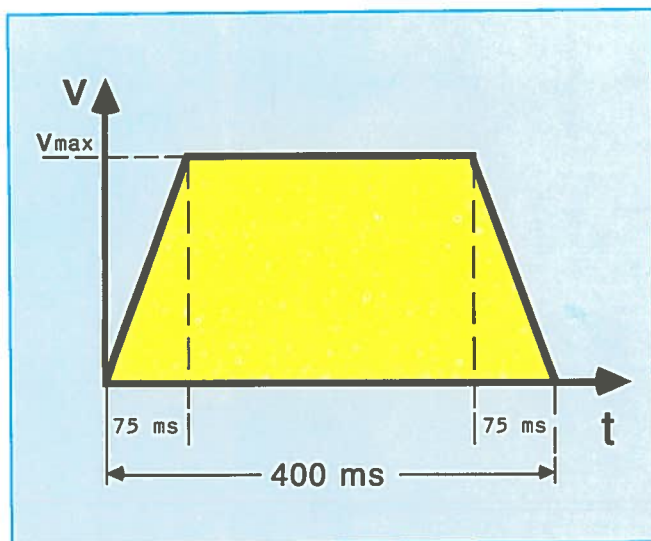


Fig. 11

$$\begin{aligned} v_{\max} &= s / (T_{\text{ges}} - T_B) \\ v_{\max} &= 80 \text{ (mm)} / (0.4 \text{ (s)} - 0.075 \text{ (s)}) \\ v_{\max} &= 246 \text{ (mm/s)} \end{aligned}$$

#### Required Flow

$$Q = A \cdot v = 9.45 \text{ (cm}^2\text{)} \cdot 24.6 \text{ (cm/s)} = 232.5 \quad [\text{cm}^3/\text{s}]$$

$$Q = 13.9 [\text{L/min}]$$

Selected:

Servo valve with  $Q_N = 20 \text{ l/min}$  at  $\Delta p = 70 \text{ bar}$

#### Pressure Drop at Valve

$$\Delta p = (Q/Q_N)^2 \cdot 70 \text{ (bar)} = (14/20)^2 \cdot 70 = 34 \quad [\text{bar}]$$

#### Acceleration

$$a_{\max} = v_{\max} / T_B = 0.25 \text{ (m/s)} / 0.075 \text{ (s)} = 3.3 \quad [\text{m/s}^2]$$

#### Acceleration\_Force

$$F_B = m \cdot a_{\max} = 500 \text{ (kg)} \cdot 3.3 \text{ (m/s}^2\text{)} = 1650 \quad [\text{N}]$$

#### Required Acceleration Pressure

$$\Delta p_{B\max} = F_B / A_R = 1650 \text{ (N)} / 9.45 \text{ (cm}^2\text{)} = 17.4 \quad [\text{bar}]$$

#### Pressure Requirements for Stopping Force

$$\Delta p_H = 6000 \text{ (N)} / 9.45 \text{ (cm}^2\text{)} = 64 \quad [\text{bar}]$$

#### Calculating the System Pressure

(see "Design criteria for open loop control with proportional valves", Page E20)

$$p_P = 2 \cdot m \cdot v / (T_B \cdot 10 \cdot A_W) + \Delta p_v + (F_{St} + F_R) / (10 \cdot A_W)$$

$$p_P = 2 \cdot 500 \text{ (kg)} \cdot 0.25 \text{ (m/s)} / (0.075 \text{ (s)} \cdot 10 \cdot 9.45 \text{ (cm}^2\text{)}) + 10 \text{ (bar)} + 6000 \text{ (N)} / (10 \cdot 9.45 \text{ (cm}^2\text{)})$$

$$p_P = 109 \text{ (bar)}$$

**p<sub>P</sub> selected 110 bar**

# The Influences of the Dynamic Characteristics of the Servo Valve on the Closed Loop Control Circuit

## Determining the Positioning Accuracy

### Loop Gain

$$K_V = K_1 \cdot K_2 \cdot K_3 \cdot K_4 = 67 \quad [s^{-1}]$$

$K_1$ = Electrical amplification		(still not known)
$K_2$ = 20 l/min/10 Volt	= 33	[cm <sup>3</sup> /s/Volt]
$K_3$ = 1/9.45 cm <sup>2</sup>	= 0.106	[1/cm <sup>2</sup> ]
$K_4$ = 10 Volt/10 cm	= 1	[Volt/cm]

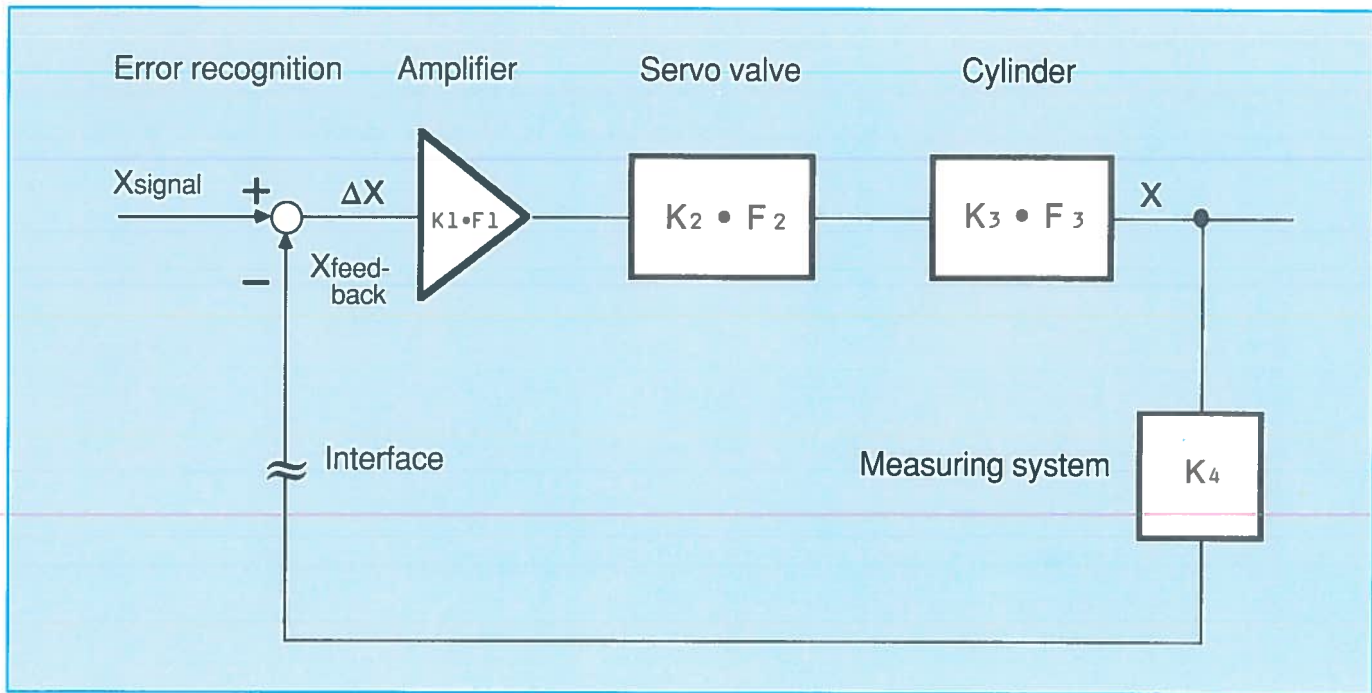


Fig. 12

### Calculation of $K_1$

$$K_1 = K_V / (K_2 \cdot K_3 \cdot K_4) = 67 \text{ (cm/s/cm)} / (33 \text{ (cm}^3\text{/s/Volt)} \cdot 0.106 \text{ (1/cm}^2\text{)} \cdot 1 \text{ (Volt/cm)})$$

$$K_1 = 19$$

### Following Error

$$s_N = v_{\max} / K_V$$

$v_{\max}$  is the maximum possible velocity when the valve is open.

$$s_N = 250 \text{ (mm/s)} / (67 \text{ (s}^{-1}\text{)}) = 3.7 \text{ mm}$$

### Positioning Accuracy

$$\Delta x \leq 5\% \text{ of } s_N$$

$$\Delta x \leq 0.19 \text{ mm}$$

### Error resulting from the reversal range of the servo valve

How large must the error be for the servo valve to overcome its reversal range?

Assumption:  $K_U = 0.2\%$  of rated signal

$$\Delta x = K_U / (K_1 \cdot K_4)$$

$$\Delta x = 0.002 \cdot 10 \text{ (V)} / (19 \cdot 1 \text{ (Volt/cm)}) = 0.001 \text{ cm}$$

$$\Delta x = 0.01 \text{ mm}$$

### Error through Change in Load

at  $F = +3000 \text{ N}$

To compensate this change in load, the servo valve must open by a certain amount, this is caused by a closed loop error  $\Delta x$ .

$$\Delta x = \Delta F / (K_1 \cdot K_2 \cdot K_3 \cdot K_4)$$

$K_2$  is the pressure amplification of the servo valve.

At 1 % signal, 80 % of the pressure is applied at the load

$$0.8 \cdot 100 / 0.1 \text{ (Volt)} = 800 \text{ bar / Volt}$$

The load error is therefore

$$\Delta s = 3000 \text{ (N)} / (18 \cdot 8000 \text{ (N/cm}^2\text{/Volt)} \cdot 9.45 \text{ (cm}^2\text{)} \cdot 1 \text{ (Volt/cm)})$$

$$\Delta s = 0.0022 \text{ cm} = 0.022 \text{ mm}$$

Notes

KUNNY ENTERPRISES (PVT)  
411 Panorama Centre  
Bldg. No. 2 Raja Ghazanfar Ali Khan Road,  
SADDAR KARACHI

Notes

Chapter K

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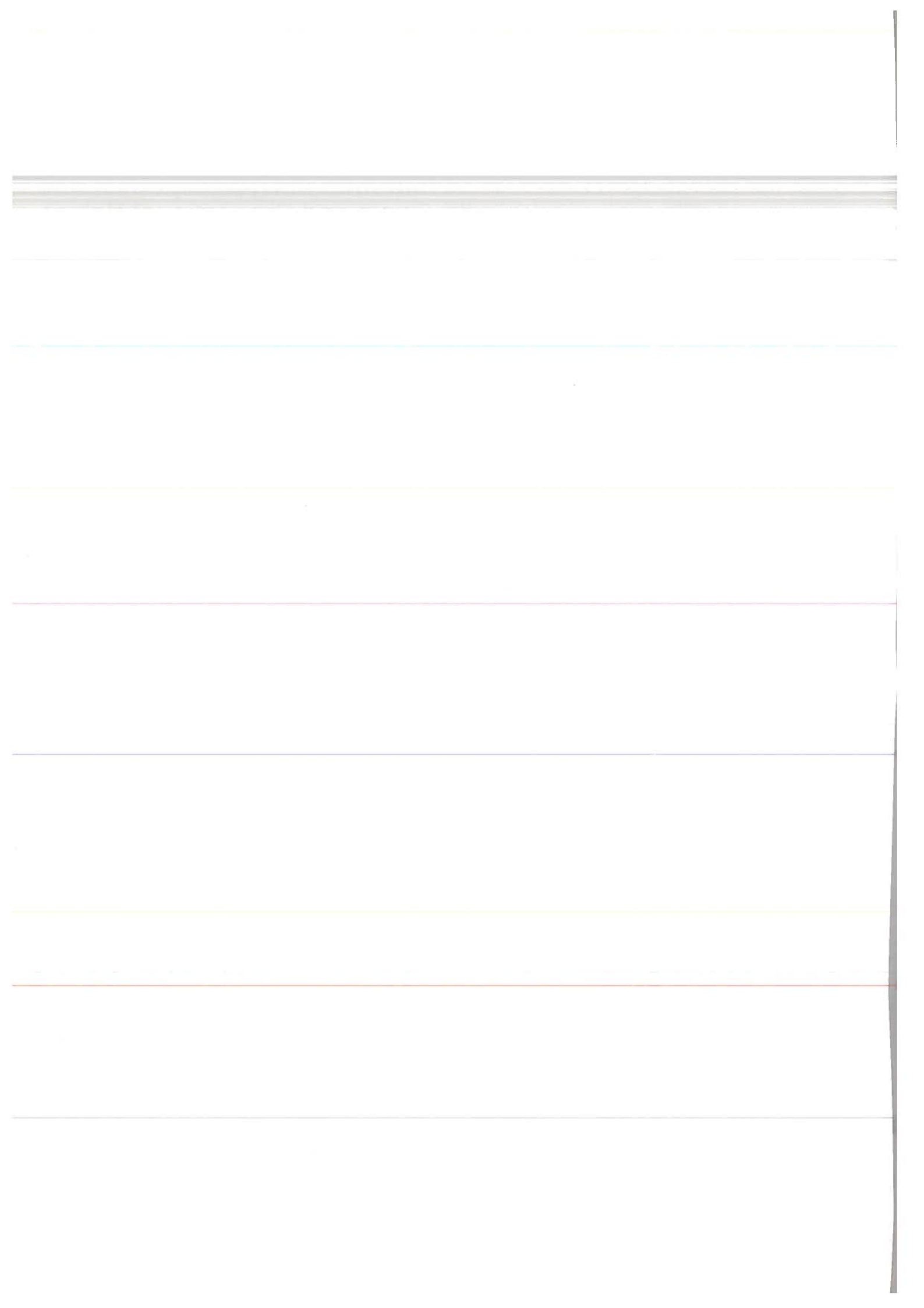
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## **Filtration in Hydraulic Systems with Servo and Proportional Valves**

Martin Reik

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## Filtration of Hydraulic Oils

Demands for increased efficiency, reduction of susceptibility to faults, longer service life, as well as making servo and proportional valves easy to service have led to valve manufacturers and operators demanding improved filtration of the hydraulic fluid.

Due to constant increases in performance of hydraulic devices, ever higher requirements are placed upon the control accuracy of valves. One of the ways to meet these requirements was to reduce the clearances between the housing and spool even further.

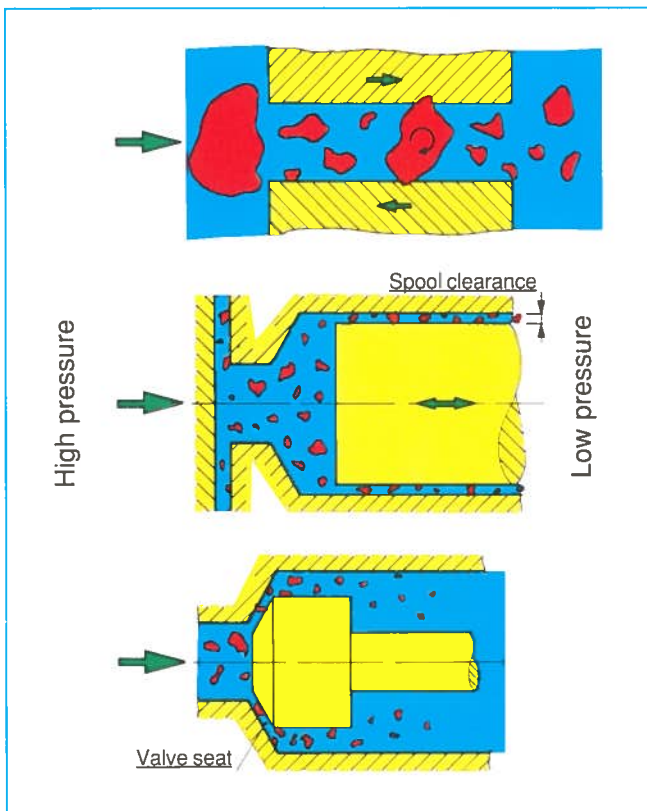


Fig. 1 The wear process, influence of dirt particles on the valve seat and spool clearance

### Effect of Contamination by Solids

#### General

Dirt particles which are much larger than the fit tolerance do not impair the valve. Particles which are smaller than the fit tolerance pass through the clearance and also do not impair the valve.

Dirt particles which are the same size as the clearance tolerance are critical for the surface of the valve and spool. The scouring effect of these dirt particles during operation results in new particles (of valve material). The particles which are larger than the valve clearance are reduced in size by the spool movement or the flow of the hydraulic medium.

The results are: Increased leakage, spool jamming, increased switching times, valve failure, change of valve characteristics.

Without suitable filtration, a chain reaction occurs which results in increased dirt concentration. After a time, these dirt particles can block the orifice in the pilot circuit.

### Effect of Erosion on Control Lands

Dirt increases the load on the material at the sensitive lands.

Result: Increased erosion due to flow resulting in inaccurate switching and control of servo and proportional valves (wear increases progressively).

Contamination entering the system from the outside can trigger off or accelerate this development. The chain reaction of particle development and increase in particle count must be minimized or even prevented by using efficient system filters.

Correct filter design and selection are reflected in higher efficiency of the overall system (reduction of downtimes) and service requirements.

The selected filter system must ensure:

- function and service life of the valves
- no sudden failure of the valves
- performance does not drop gradually due to increasing internal leakage
- that the valve setting data does not change over the period of operation
- that there is no change in the valve characteristics, e.g. due to jammed dirt particles.

All too often, the hydraulic filter is neglected or even forgotten in the planning of a hydraulic system. The filter is then often quickly installed during assembly of the system.

For reasons of cost or space, a filter is then selected which is often too small and too coarse. The system operator is then faced with difficulties resulting from short service life of the filter elements (filter too small) or frequent failures of servo and proportional valves (filter too coarse) so that considerable additional costs are incurred.

### Contamination by Solid Particles in Hydraulic Systems

A differentiation is made between the following types of contamination:

#### Initial Contamination

This type of contamination of the hydraulic oil takes place during installation and commissioning of hydraulic systems (dust, scale, metal swarf, weld beads, fluff, rust, residual packing, paint particles etc.).

Contamination During Operation

Entry of dirt at the hydraulic tank through poor tank breathing, pipe inlet points, piston rod seals etc. The contamination rate greatly depends on the relevant application, e.g. quarries, road construction, cement works etc.

Contamination Through Fresh Oil

Fresh oil delivered by the oil supplier often has an impermissibly high contamination level for servo and proportional valves. This contamination must be removed by filters installed in the system.

In systems which only have one return line filter, however, the "first fill" contamination can result in serious damage to the components even when flushing the system. It is therefore necessary to fill the system with fresh oil or to change the oil using an oil service unit, or via the return line filter already fitted.

The hydraulic filter used in the service unit must have the same pore size of the filter element as the filter in the hydraulic system.



Fig. 2 Oil service unit

Fig. 3 Comparison table purity classes

Contamination Classes for Hydraulic Oils

The contamination classes indicate how many particles of a certain particle size are contained in 100 ml of hydraulic fluid.

The contamination class is determined by counting and "grading" of the dirt particles. This takes place either with the aid of a microscope or with electronic particle counters. In contrast to particle counting with a microscope, counting with the electronic particle counter is subject to far less emotional conditions. As of a dirt concentration of approx. 10 mg per liter or excessive turbidity of the fluid, the contamination can only be determined by weight (gravimetric analysis). However, the individual dirt particles cannot be classified using this method.

The servo and proportional valves are normally the components most sensitive to dirt in a hydraulic system. They therefore determine the overall contamination class of the hydraulic oil and also the necessary pore size of the filter element.

Contamination Classes

Presently, 5 classification systems (ISO 4406 or CETOP RP 74H, NAS 1638, SAE, Mil. std. 1246 A) are available.

As can be seen from the following table, the systems are comparable.

ISO 4406 or Cetop RP 70 H	Particles per ml > 10 µm	ACFTD Solid content mg/l	MIL STD 1246 A (1967)	NAS 1638 (1964)	SAE (1963)
26/23	140 000	1 000			
25/23	85 000		1 000		
23/20	14 000	100	700		
21/18	4 500			12	
20/18	2 400		500		
20/17	2 300			11	
20/16	1 400	10			
19/16	1 200			10	
18/15	580			9	6
17/14	280		300	8	5
16/13	140	1		7	4
15/12	70			6	3
14/12	40		200		
14/11	35			5	2
13/10	14	0,1		4	1
12/ 9	9			3	0
18/ 8	5			2	
10/ 8	3		100		
10/ 7	2,3			1	
10/ 6	1,4	0,01			
9/ 6	1,2			0	
8/ 5	0,6			00	
7/ 5	0,3		50		
6/ 3	0,14	0,001			
5/ 2	0,04		25		
2/ 8	0,01		10		

### The ISO 4406 System

In the diagram, the particle sizes are specified on the X axis. The particle count is entered on the Y axis and divided into the classification numbers 1 – 20. The straight lines entered in the diagram describe the particle distribution in the hydraulic oil. The slope of the straight lines is determined by entry of the particle sizes of 5  $\mu\text{m}$  and 15  $\mu\text{m}$ . The straight particle distribution curve is described by determining the classification number at 5  $\mu\text{m}$  particles and 15  $\mu\text{m}$  particles (see Fig. 4).

The following oil purity is necessary for servo and proportional valves:

Servo valves                    13/10 (red curve)  
Proportional valves        17/14 (blue curve)

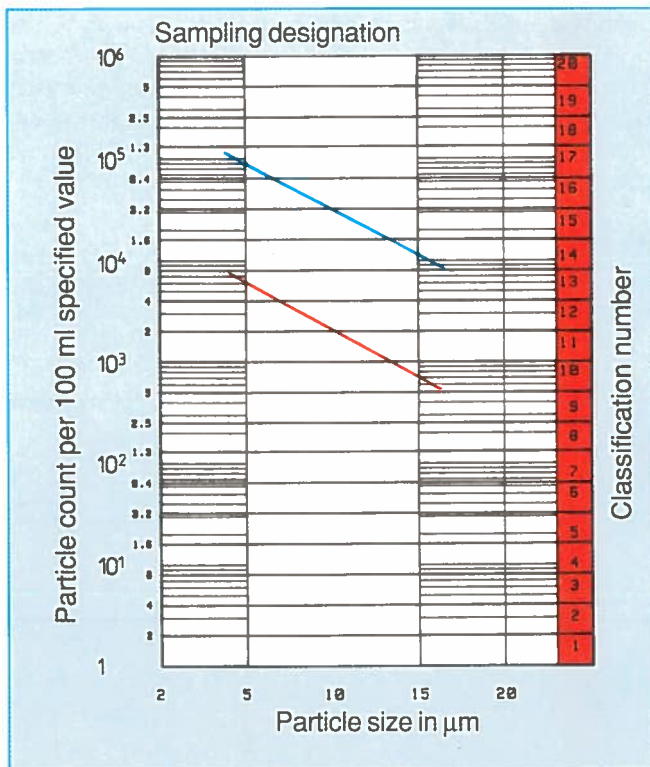


Fig. 4 Structure of purity class to ISO 4406 or Cetop RP 70 H

### NAS 1638 System

The individual particle sizes are combined in 5 ranges. A maximum particle count is specified in each class for each range (see Fig. 5).

Necessary oil purity:

Servo valves                    NAS 4 to 6 (red range)  
Proportional valves        NAS 8 to 9 (blue range)

### SAE Contamination Class

Due to the relatively low number of contamination classes (9 particles/ml to 580 particles/ml), this method of contamination classification is rarely used.

### Advantages and Disadvantages of NAS 1638 and ISO 4406

#### Advantage of NAS 1638:

Counted particles can be immediately assigned to a class.

#### Disadvantage of NAS 1638:

No precise description of the actual particle distribution. The required class can only be maintained in one particle size range. For this reason, the defined particle size range must be specified during classification to NAS 1638.

#### Advantage of ISO 4406:

Description of actual particle distribution. Any contamination status of the hydraulic fluid can be described.

#### Disadvantage of ISO 4406:

The evaluation method requires a great deal of time. Measured particle count must firstly be converted into an ordinal number, thereby permitting description of the straight distribution curves.

Maximum number of dirt particles in 100 ml hydraulic fluid at particle size														
$\mu\text{m}$	class													
	00	0	1	2	3	4	5	6	7	8	9	10	11	12
5-15	125	250	500	1000	2000	4000	8000	16000	32000	64000	128000	256000	512000	1024000
15-25	22	44	89	178	356	712	1425	2850	5700	11400	22800	45600	91200	182400
25-50	4	8	16	32	63	126	253	506	1012	2025	4050	8100	16200	32400
50-100	1	2	3	6	11	22	45	90	180	360	720	1440	2880	5760
> 100	0	0	1	1	2	4	8	16	32	64	128	256	512	1024

Fig. 5 Structure of purity class to NAS 1638

### Sampling Hydraulic Fluids

#### General

- Before taking a sample, the measuring equipment must be thoroughly flushed with a suitable solvent.
- Only use sampling bottles cleaned with a little clean solvent.
- Remove any remaining solvent before taking a sample.
- Sampling volume: min. 250 ml.
- Before taking the actual sample, flush the sampling equipment with 2 l of system fluid.
- Take 0-sample (this is not used for analysis)
- Fill the fluid to be examined in a new, cleaned sampling flask. The sampling unit must penetrate the protective film (do not remove film from sampling bottle).

#### Types of Sampling

- Dynamic sampling  
Sampling point: Systems in operation (turbulent flow must be ensured). Refer to ISO 4021.
- Static sampling  
Sampling point: From the hydraulic tank (stationary system). Refer to CETOP RP 95 H, Section 3.

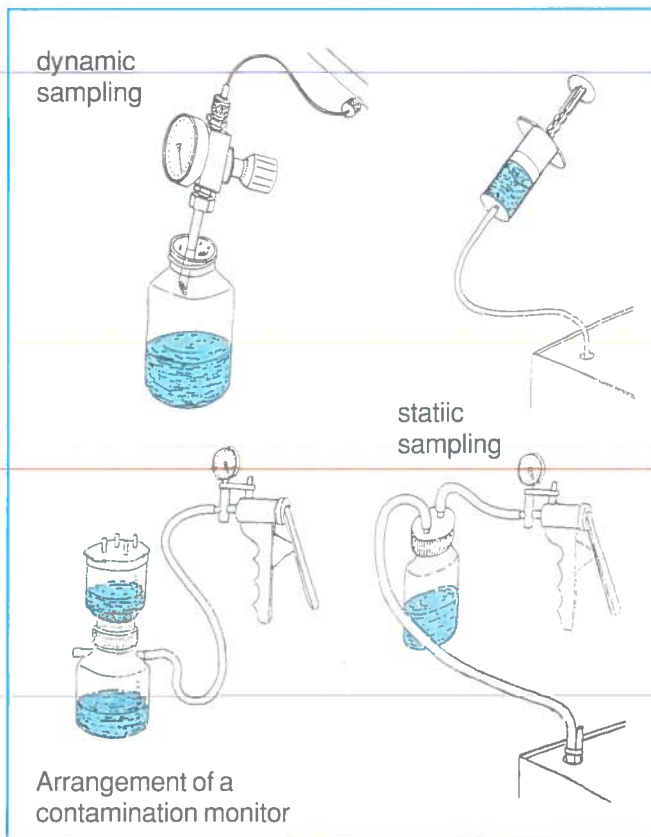


Fig. 6 Types of hydraulic fluid sampling

#### Advantages and Disadvantages of the Types of Sampling

- Advantage of dynamic sampling:  
The oil quality can be measured directly after the filter or after the valve. This enables precise definition of the amount of dirt fed to the valve.
- Disadvantage of dynamic sampling:  
The sampling points must be provided as early as the system planning stage, all special adaptors must be made. Complicated sampling equipment.
- Advantage of static sampling:  
Extremely easy sampling from hydraulic tank.
- Disadvantage of static sampling:  
Only the oil quality in the hydraulic tank is determined, not directly at the valve.

The selection of the sampling point can lead to incorrect definition of the oil purity, e.g. a sample taken from the bottom of the tank will indicate a different degree of oil contamination than a sample taken at the surface.

#### Multipass Test to ISO 4572

With the aid of this test, the separation ratio and the dirt holding capacity of the filter element is determined.



Fig. 7 Multipass test bench

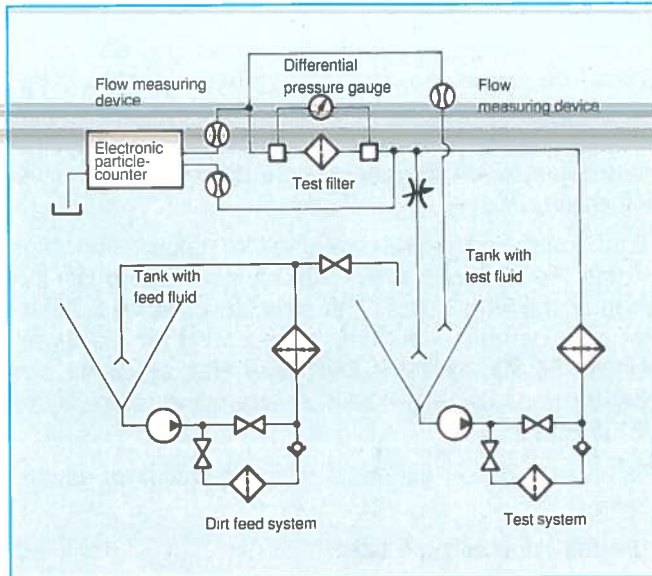


Fig. 8 Circuit diagram of the test bench to ISO 4572

2 hydraulic circuits are built up on the test bench.

The test system with tank, test medium, pump, cooler/heater, flow measuring device, filter with test element and electronic particle counters.

The feed circuit with pump, cooler/heater, injection nozzle and with feed fluid. The fluid in this tank is contaminated with test dirt (ACFTD).

Before carrying out the test, both systems are cleaned with ultrafine filters. The test is started only when the specified number of dirt particles are in the test circuits.

### Test Sequence

The feed circuit constantly feeds a small quantity of fluid into the main circuit.

The contaminated test fluid is now fed to the filter element. Fluid tests are taken upstream and downstream of the test filter and counted in the electronic particle counter. At the same time, the differential pressure is measured which results from contamination of the filter element.

The  $\beta_x$  value acts as the measure for the retention rate at the filter element.

### Calculating the $\beta_x$ Value

The dirt particles counted before the filter element of a certain particle size  $x$  are divided by the dirt particles counted after the filter element (same particle size  $x$ , at same differential pressure and at the same time). The dimensionless number obtained represents the  $\beta_x$  value.

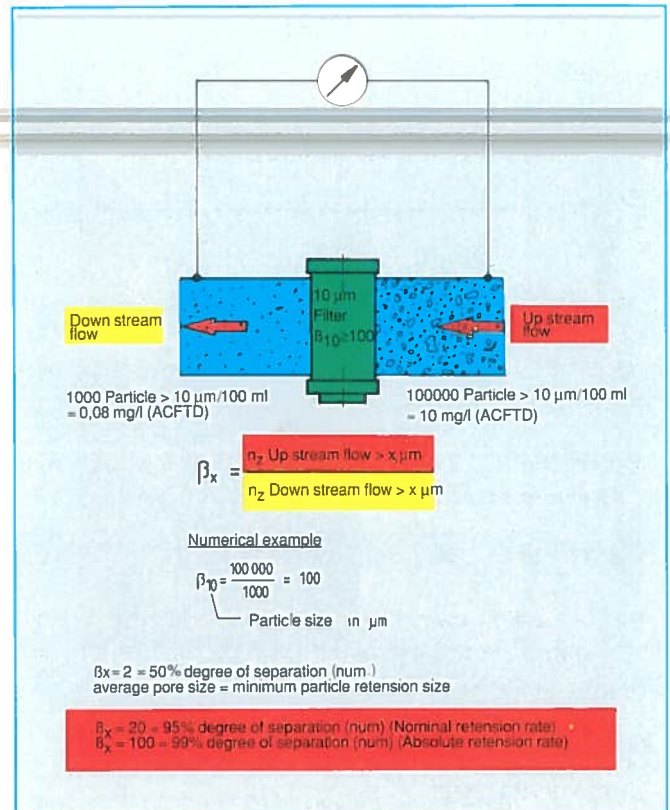


Fig. 9 Separation of dirt particles by filter elements

### Numerical Example

Measured particle count:

Upstream flow: 10000 particles > 3 µm in 100 ml  
Downstream flow: 100 particles > 3 µm in 100 ml.  
 $\beta_3 = n_z \text{ upstream flow} / n_z \text{ downstream flow} = 10000/100 = 100$   
 $\beta_3 = 100 = 99 \% \text{ separation (also termed degree of separation)}$

The  $\beta_x$ -value describes the separation characteristic (degree of efficiency) of a filter element. The advantage lies in the fact that the range between 90 % and 100 % degree of separation can be considerably wide spread.

Dimensionless  $\beta_x$ -values can be converted at any time into a %-value of the degree of separation (see Fig. 10).

### Why use $\beta_x$ -values?

Formerly, filtration data was based on various in-company tests carried out by the various filter manufacturers. Specifying the  $\beta_x$ -value while taking into consideration the resulting differential pressure makes it possible to compare the filter element pore size of various filter manufacturers.

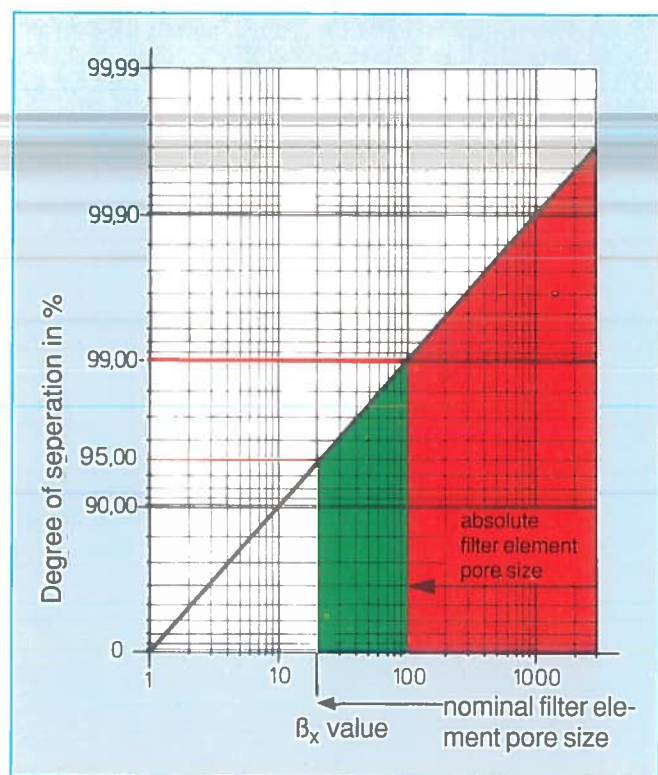


Fig. 10  $\beta_x$ -value as a function of the degree of separation in %

## Definition of Filter Element Pore Size

Before the  $\beta_x$  value was established, no comparative data could be given for filter pore size. A clear definition became possible with the introduction of the  $\beta_x$  values.

There were two definitions of filter pore size:

**Nominal filtration:** - In this case, useful  $\beta_x$  values are not defined. This means for the user, that only part of the filterable contamination is filtered out to the pore size stated.

**Definition:**  $\beta_x \leq 20$ . Corresponds to a degree of separation of approx. 95%.

**Absolute filtration** As of a  $\beta_x$ -value of 100 or a degree of separation of 99 %, the filter element pore size is referred to as the absolute retention rate.

## Characteristic Features of Filter Elements with Multi-layer Matt Constructions

(e.g. Rexroth - and Hydac filter elements - Betamicon)

Experience gained in particular applications and test laboratories has led to the development of filter elements of multi-layer matt construction.

Tests and analysis have also shown that the degree of oil purity required by the manufacturers of servo and proportional valves can only be met with this matt filter design.

The direction of flow through the element must always be from the outside towards the inside. The filter matts should be folded in a star arrangement to ensure the largest possible filter area is available in the installation chamber of the filter element. The design of the filter matt depends on the permissible differential pressure of the filter.

High quality adhesives are used to secure the filter matt in the end caps of the filter element and to join the ends of the filter matts. The strength of these adhesives considerably decreases at operating temperatures above 100 °C, meaning that these filter elements can only be used up to a maximum operating temperature of 100 °C.

These Betamicon elements offer the following advantages:

- precisely defined pore size,
- excellent retention of extremely fine particles over a wide differential pressure range,
- high dirt holding capacity through large specific filter area
- good resistance to chemicals
- high burst pressure strength prevents damage to filter element, e.g. during cold start, switching and differential pressure peaks
- water and water constituents in the hydraulic fluid do not have a detrimental effect on filter performance

## $\beta_x$ -Stability

$\beta_x$ -values can be specified at high differential pressures for these filter elements. As shown in Fig. 11, Betamicon filter elements of the type BH maintain constant  $\beta_x$ -values up to high differential pressures at the filter element.

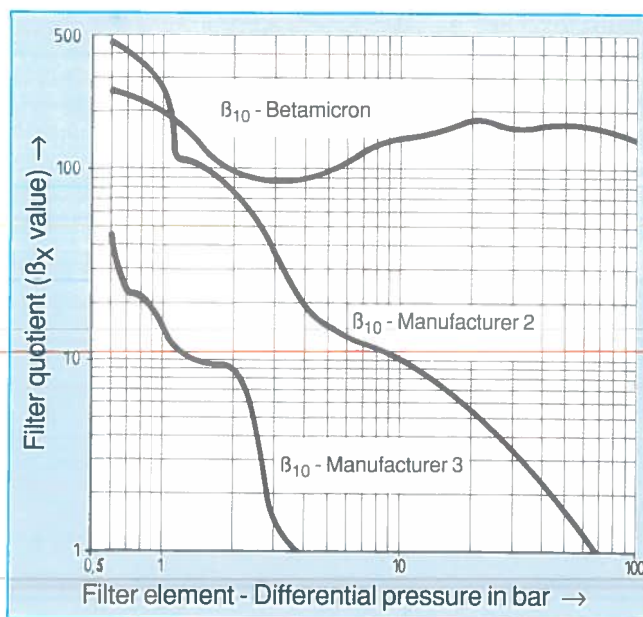


Fig. 11 Comparison of  $\beta_{10}$  values over constant differential pressure of elements of various manufacturers

This high  $\beta_x$ -value stability is necessary in order to ensure trouble-free operation of servo and proportional valves.

Dynamic-hydraulic loads, pressure peaks resulting from rapid switching procedures, shock-like changes in flow and various temperature ranges, ignoring the clogging indicator, do not impair the retention properties of these filter elements.

In the case of return line filter elements, with built-in bypass filter,  $\beta_x$  values must be maintained up to a differential pressure which is a multiple higher than the opening pressure of the bypass valve or the response point of the clogging indicator.

### Design Features of Betamicon Filter Elements

**Direction of flow:** From outside to inside. Flow in the opposite direction will destroy the filter element (negative pressure peaks, distorted filter elements).

In these cases, fast closing non-return valves must be installed after the filter element. The installation of filter housings with integrated non-return valves (e.g. filter type DFF) has a proven record of success in such cases.

**Corrugated arrangement:** The filter elements feature a corrugated filter matt (matrix) to provide the largest possible filter area and a long service life of the filter element.

**Service life of filter element:** The service life or the change interval of a filter element is determined by its

dirt holding capacity. This can vary considerably for the same element under different operating conditions.

The influencing variables are:

- contamination of the system,
- hydraulic load of the element,
- effective differential pressure spectrum at the filter element.

The contamination of the system is determined by the dirt production of the system, the dirt penetration rate, the particle size and particle count, as well as the type of contamination.

The influencing variables for the hydraulic load are filter area, flow, viscosity, operating pressure and operating medium.

The influencing variables for the filter elements are defined by an effective particle retention size, the specific dirt holding capacity and the design of the filter element matt.

In order to make available the largest possible effective differential pressure spectrum for filtration, definition of the filter size should be based on the lowest possible pressure loss with a clean filter element. This is illustrated in Fig. 12 where the differential pressure at the filter element is shown at increasing contamination and operating time. It can be clearly seen that at a low initial  $\Delta p$ , a higher real dirt holding capacity is possible than at a high initial  $\Delta p$ . In both cases, the bypass valve, maintenance indicator or differential pressure strength of the filter element produce the upper limit for the maximum filter element load.

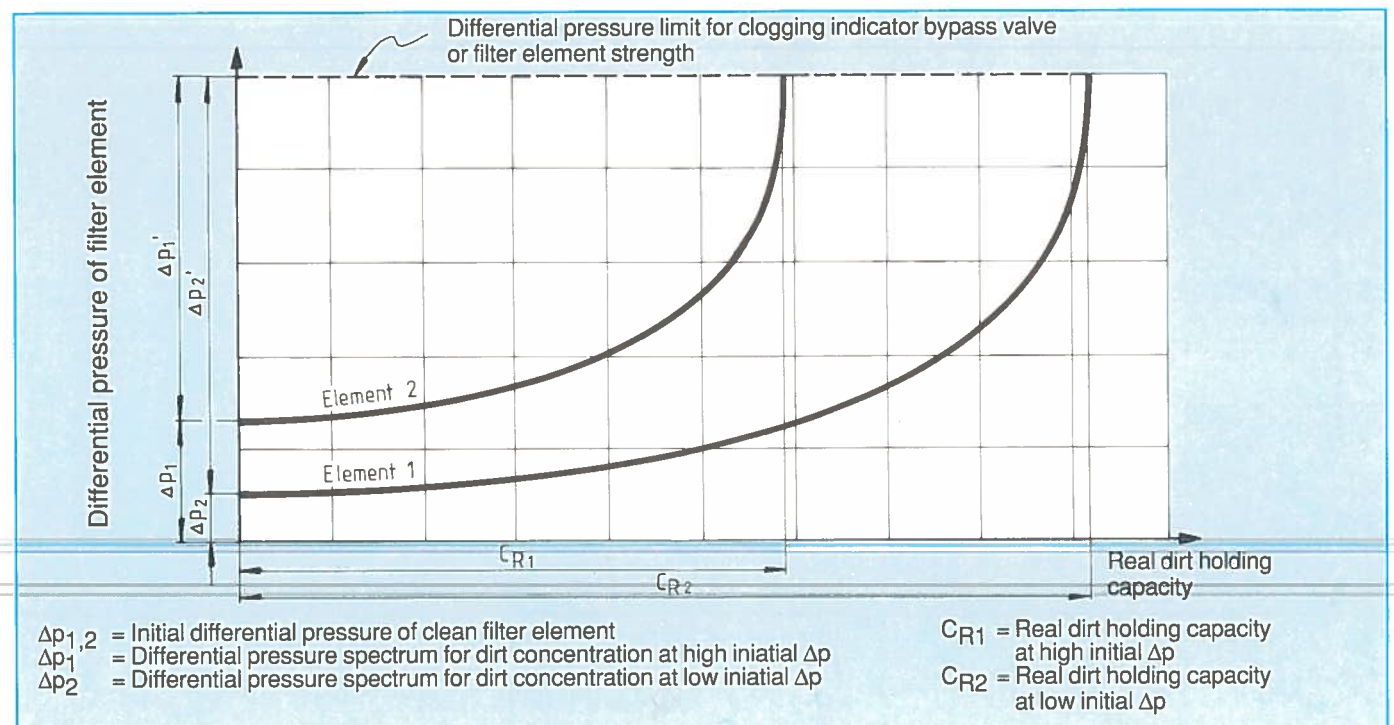


Fig. 12 Retention capacity dependent on initial  $\Delta p$

## Filter Design

In addition to requirements with regard to functional reliability and service life of servo and proportional valves, plant and operating costs also represent a decisive factor when determining the most suitable hydraulic filter.

Considerable improvements can be achieved in the operational reliability and service life of servo and proportional valves by selecting fine pore, high quality filter elements.

The following criteria must be taken into consideration when determining the filter size, the filter element pore size and the type of filter:

- Dirt sensitivity of the servo and proportional valves:  
Note filter element pore size or required purity class

- Application of the overall hydraulic system  
The contamination from the environment must be taken into consideration (laboratory system or smelting plant).

- Determination of flow  
Flow can at times be greater than the maximum pump delivery (double rod cylinders or return lines consisting of several circuits).

- Permissible pressure drop (housing and element) for clean filter element

**For pressure line filters:** 1.0 bar, with clean filter element and operating viscosity

**For return line filters:** 0.5 bar, with clean filter element and operating viscosity

- Permissible differential pressure of the filter element must correspond to system conditions at the filter installation point.

- Compatibility of the filter materials with the pressure medium must be ensured.

- Design pressure of the filter housing (operating pressure)

- Determination of filter type:  
Which type of clogging indicator should be installed (visual, electrical, electronic). Bypass filters must not be installed in the case of pressure line filters.

- Operating temperature or design temperature

## Filter Arrangement in Hydraulic System

### Basis of Design

The pore size of the filter element selected corresponding to the relevant application should be the same in all filters used in the hydraulic circuit (hydraulic system - and filter-breather).

In the case of systems with high oil volume, main flow filtration is normally carried out with a return line filter (filter pore size 20 µm absolute). The necessary purity class for the operating medium in conjunction with proportional and servo valves is achieved by using a pressure line filter with the necessary pore size immediately before the valve.

In addition to this arrangement, it is also recommended to install a bypass filter system with a filter element pore size of 5 µm absolute.

Caution: The pressure-line-filter should be larger in this filter arrangement due to the higher contamination which can be expected.

Contamination classes				Proposed pore size $\beta_x \geq 100$	Application
Cetop RP 70 > 5 µm	RP 70 > 15 µm	NAS 1638 > 5 µm	NAS 1638 > 15 µm		
13	10	4 u. 5	3 u. 4	X = 3	Servo valves at an operating pressure > 160 bar
15	12	6 u. 7	5 u. 6	X = 5	Servo valves at an operating pressure < 160 bar
17	14	8 u. 9	7 u. 8	X = 10	Proportional valves

Element type	Filter element strength	Element pore size x	Rexroth element designation	Applications
BH/HC	210 bar	3	... D 003 BH/HC	<b>Pressure line filter</b> Safeguarding function and service life of servo and proportional valves
		5	... D 005 BH/HC	
		10	... D 010 BH/HC	
BN/HC	30 bar	3	... R 003 BN/HC	<b>Return line filter</b> with bypass valve opening pressure 3 bar
		5	... R 005 BN/HC	
		10	... R 010 BN/HC	
		3	... D 003 BN/HC	<b>Bypass filter</b> filter element for flushing a system
		5	... D 005 BN/HC	
		10	... D 010 BN/HC	

The maximum operating temperature of the filter elements is 100 °C.

Filter elements must be subject to quality inspection during manufacture (ISO 2942).

### Selection of Filter Housing

Filter type	Determination of filter size	Remarks
Pressure line filter	$\Delta p_{\text{housing}} + f \cdot \Delta p_{\text{element}} \leq 1,0 \text{ bar}$	<b>without</b> Bypass valve
Return line filter	$\Delta p_{\text{housing}} + f \cdot \Delta p_{\text{element}} \leq 0,5 \text{ bar}$	<b>with</b> Bypass valve
Bypass filter	$\Delta p_{\text{housing}} + f \cdot \Delta p_{\text{element}} \leq 0,3 \text{ bar}$	Pump delivery approx. 5-10% of the tank capacity <b>without</b> bypass valve

f = Viscosity conversion factor

### The Effect of Viscosity in Filter Design

The characteristic curves for the filter housing and filter elements specified in the brochures are referred to a viscosity of 30 mm<sup>2</sup>/s for instance. If the design viscosity (normally operating viscosity) varies from this reference viscosity, then the pressure loss in the filter element (diagram specification) must be converted to the pressure loss at operating viscosity.

The viscosity conversion factor "f" is used for this conversion.

### Determination of the viscosity conversion factor "f"

$$f = \frac{\text{Design viscosity}}{30} + \sqrt{\frac{\text{Design viscosity}}{30}}$$

Panzer - Beitler calculation formula, "Arbeitsbuch der Ölhydraulik - Projektierung und Betrieb", 2nd edition 1969.

The scope of validity for this calculation formula is between 30 and 3000 mm<sup>2</sup>/s.

### Determination of Factor "f" with the Aid of a Diagram

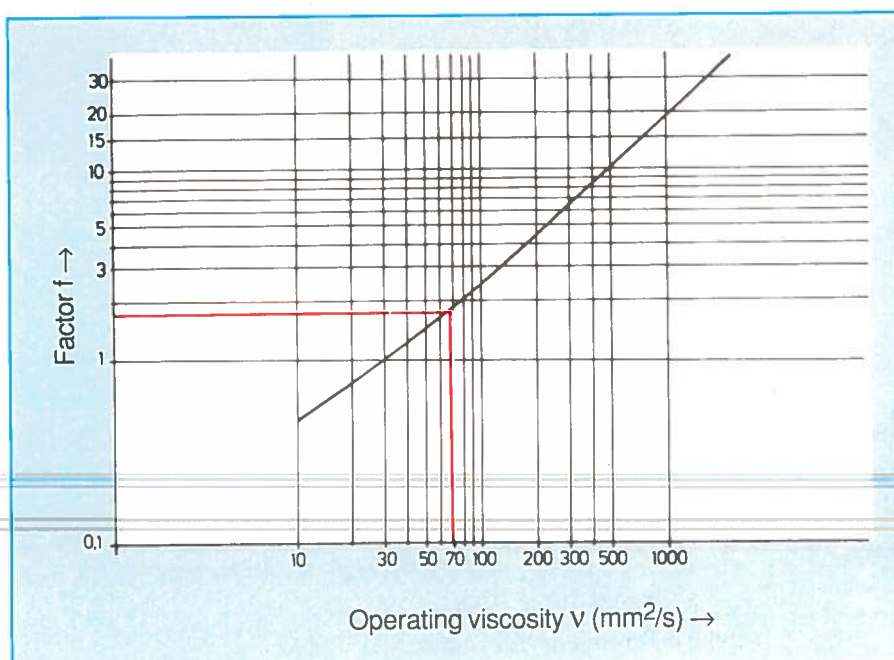


Fig. 13 Graphic representation of the viscosity conversion factor

## Example

The following curves are based on 30 mm<sup>2</sup>/s. The hydraulic system is operated with hydraulic oil ISO VG 68. The operating temperature is 40 °C. The hydraulic filter should be designed corresponding to the specified operating temperature.

1) Calculation of factor  $f$  with aid of formula:

$$f = (68/30 + \sqrt{68/30})/2 = 1.89$$

$$f = 1.89$$

2) Determination of factor  $f$  with aid of diagram:

The factor " $f$ " can be read off directly from Fig. 13 (red line).

$$f = 1.9$$

## The effect of the density of the liquid to be filtered on the filter size

The housing diagrams are based on a density of 0.86 kg/dm<sup>3</sup> (mineral oil). If this density changes, the differential pressure of the housing given, must be converted proportional to the change in density.

## Determination of Total Differential Pressure from Housing Curves and Filter Element Curves

### Example:

- Hydraulic system with proportional valves
- Flow 50 l/min
- Type of oil: ISO VG 68
- Operating temperature 40 °C
- Operating pressure 300 bar
- With electrical maintenance indicator.

### Procedure:

- Determination of filter pore size

Choose the correct pore size from the table "Selection of correct pore size",.

E.g. application area: Proportional valves. Suggested pore size 10 µm (B10 ≥ 100) can be read off.

- Determination of filter type

The filter should be installed directly before the proportional valve (functional reliability and service life). Operating pressure 300 bar: DF - filter housing without bypass filter must be used.

- Determination of filter element design

Select element type and element designation from the diagram "Selection of Filter Elements".

E.g.: Application: Proportional valves, ensuring functional reliability and service life.

Necessary type of filter element: BH/HC,

Element designation: ... D 10 BH/HC

- Determination of viscosity conversion factor " $f$ "

Determine factor  $f$  from Fig. 13:  $f = 1.9$

- Determination of filter size

Assumed size: DF ... 110

Determine the pressure loss at  $Q = 50$  l/min from housing diagram.

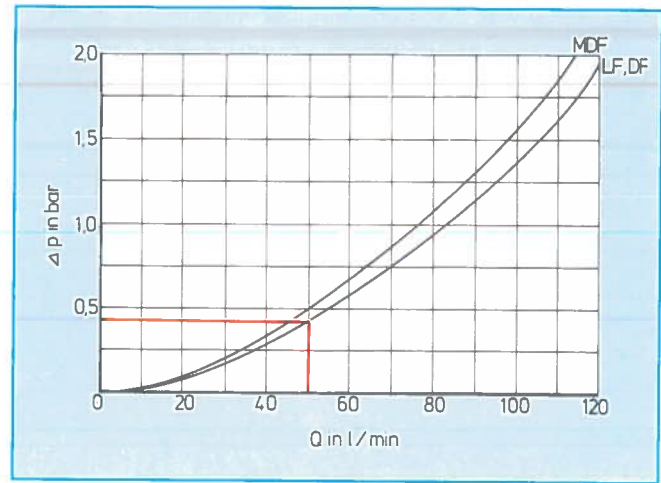


Fig. 14

Housing diagram from pressure line filter brochure

$\Delta p_{\text{housing}} = 0.4$  bar.

Determine the pressure loss at  $Q = 50$  l/min filter element diagram

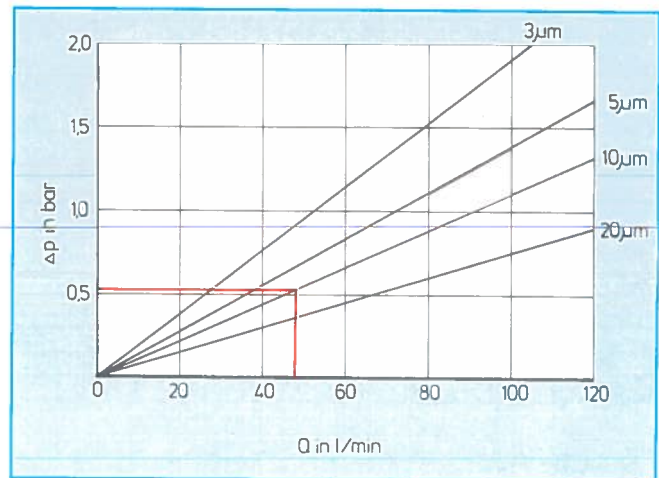


Fig. 15 Filter element diagram

$\Delta p_{\text{element}} = 0.6$  bar

- Determination of overall differential pressure

$$\Delta p_{\text{total}} = \Delta p_{\text{housing}} + f \cdot \Delta p_{\text{element}}$$

The following calculation results for DFBH/HC 110 G 10 C1.X:

$$\Delta p_{\text{total}} = 0.4 \text{ bar} + 1.9 \cdot 0.6 \text{ bar} = 1.54 \text{ bar.}$$

The total pressure difference determined in this way is higher than 1.0 bar.

This means the pressure line filter DF BH/HC 110 G 10 C 1.X is too small.

The same calculation must now be repeated with a larger filter size. The filter is the correct size when the calculated total pressure difference is below the specified maximum initial differential pressure.

Diagrams for determining the filter size are provided to simplify this relatively complicated procedure of filter design.

### Procedure with Diagrams for Determining the Filter Size

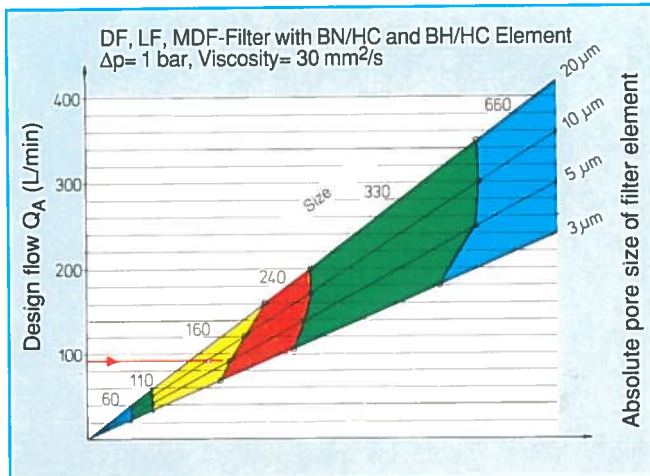


Fig. 16 Diagram for determining the filter size of pressure line filters

The filter size is determined by the point of intersection of flow and filter pore size.

If the design viscosity deviates from the viscosity of  $30 \text{ mm}^2/\text{s}$  used as the basis in the diagram, then the flow must be increased by the viscosity conversion factor "f".

$$Q_{\text{diagram}} = Q_{\text{design}} \cdot f$$

Example: Proportional valve (previous example)

$$Q_{\text{diagram}} = 50 \text{ l/min} \cdot 1.9 = 95 \text{ l/min.}$$

Determination of size from diagram (Fig. 16)

The point of intersection for flow of 95 l/min with the  $10 \mu\text{m}$  filter pore size line is in the area indicating filter size 160.

The following filter must therefore be used:

DFBH/HC 160 G 10 C 1.X

The Following Diagrams (Figs. 17 and 18) are used to Determine Return Line Filters (for Tank Installation)

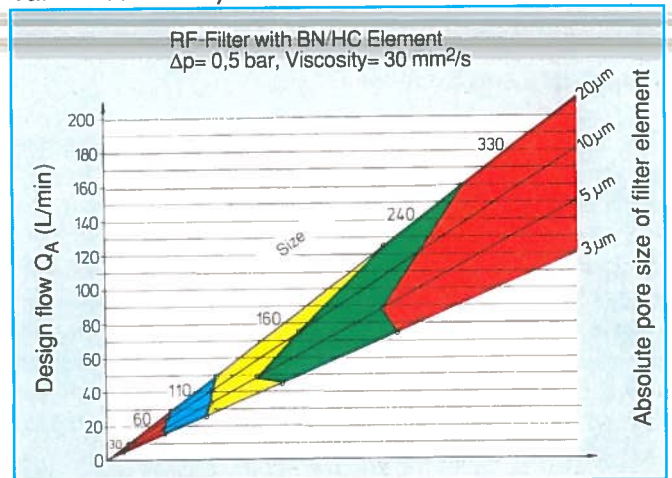


Fig. 17 Diagram for determining the filter size of return line filters Q to 200 l/min

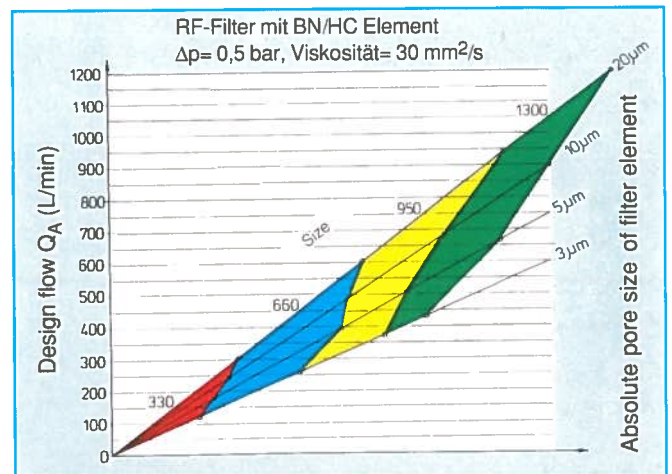


Fig. 18 Diagram for determining the filter size of return line filters Q to 1200 l/min

### Design of Filters for Filtration of Fire resistant Fluids

Standard filters only be used to a limited extent for filtration of fire resistant fluids.

Changes must be made to the filter housing or filter element to suit the fluid being used.

Regarding the filtration of these fluids, particular attention should be paid to the compatibility of the media with the filter materials. Today, filter manufacturers have gained sufficient experience to be able to offer suitably resistant filters corresponding to the type of fluid. This involves the use of different materials or the implementation of certain surface protection measures. This also applies to clogging indicators and other accessories. In addition, compared to the use in mineral oil, it is advisable to use filters of larger design. This is necessary due to the higher wear characteristics of

the components, the soap residue, the formation of micro organisms, as well as the different dirt binding capacity. Cooperation with the filter manufacturer is recommended, when designing the filter system.

### Design of Tank Breather Filters

The dirt penetration rate has a considerable influence on the contamination of a system. Tank breathing is of particular importance in this connection. Its function is to prevent ambient contamination entering the system despite air exchange. An incorrectly or poorly designed tank breather can result in considerable additional load on the filter circuit and therefore in short service life of the filter elements. The performance values of the breather filters should be adapted to those of the system filters. The following data must be taken into consideration when designing the breather filter:

Pore size of filter element:  $\beta_3 \geq 100$

Design volume for the air filter:

10 times the maximum volume fluctuation in the fluid tank.

Design differential pressure with clean filter element; and design volume: 0.02 bar.

### Design Features of Hydraulic Filters

#### Pressure Line Filter (For installation in line)

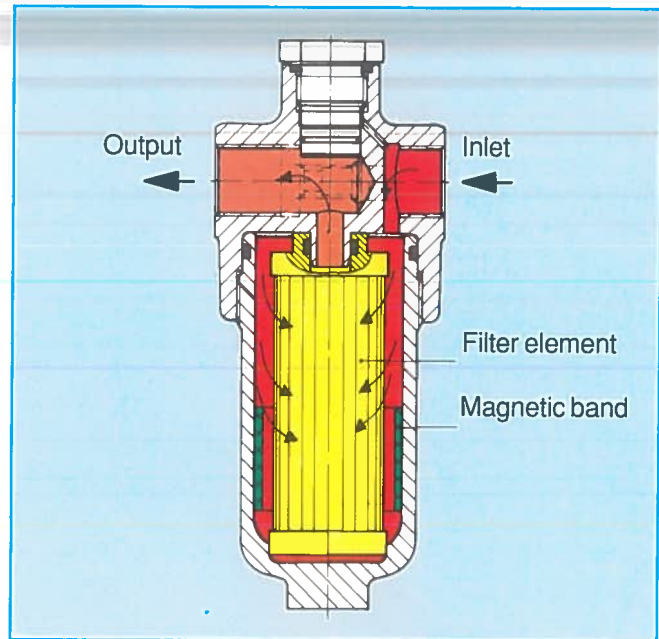


Fig. 19 Sectional view of a pressure line filter

These filters should be used without bypass valves. The flow through the filter element must always be from the outside towards the inside (note direction of flow indicated by arrow on filter head).

A filter clogging indicator must be used.

#### Pressure Line Filter for Direct Attachment to Proportional and Servo Valves

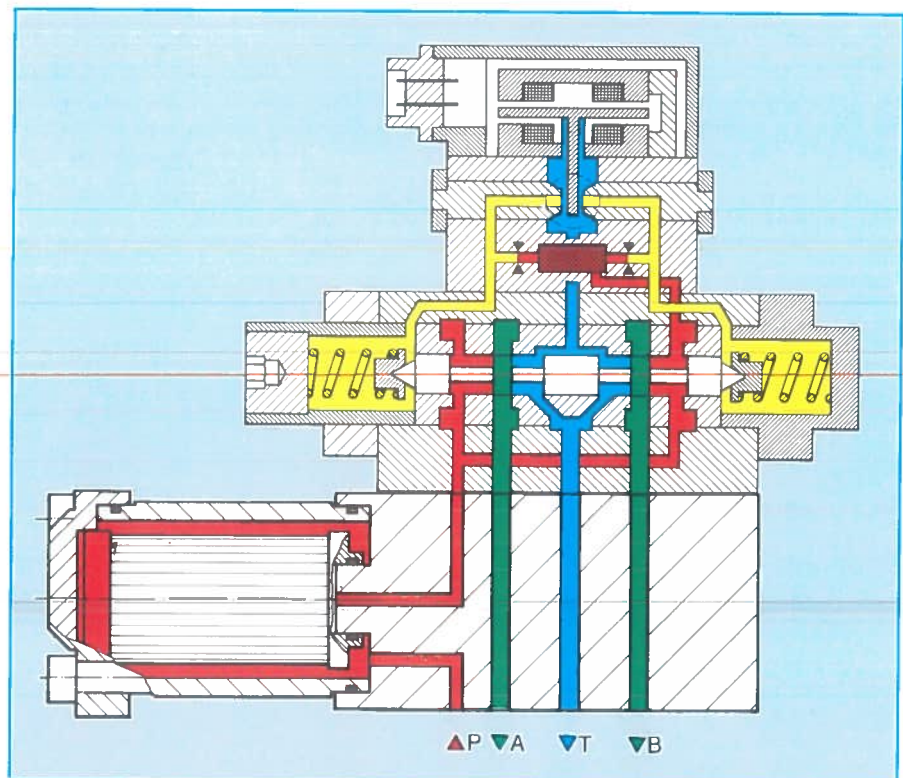


Fig. 20 Pressure line filter, arranged directly beneath the valve

This filter arrangement ensures that contamination of the hydraulic fluid no longer occurs between the filter and valve and that the system can be flushed with the valve functioning.

### Return Line Filter (For tank installation)

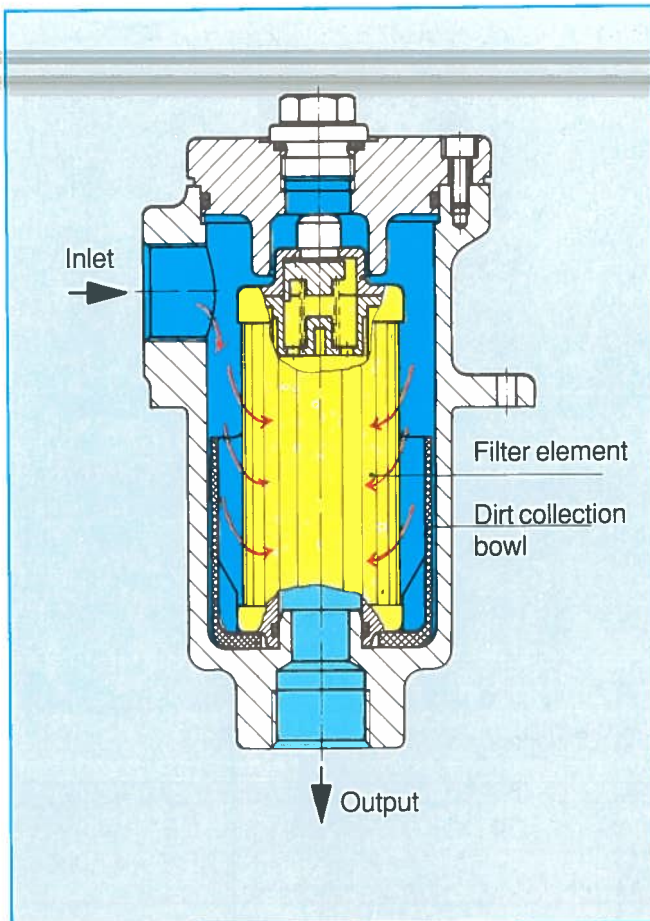


Fig. 21 Sectional view of a return line filter (tank installation)

To ensure valves or other hydraulic devices cannot operate incorrectly, return line filters are normally installed with bypass valves.

The flow of medium through the filter element normally takes place from the outside towards the inside.

A filter clogging indicator should be used otherwise opening of the bypass valve would not be registered.

The built-in dirt collection bowl prevents heavily contaminated fluid from flowing into the tank when changing the filter element.

The specified operating pressure of 25 bar is referred to the filter housing under dynamic load.

#### Clogging Indicator

Various types of clogging indicators are available to indicate and monitor the change and cleaning intervals for the filter elements. Care must be taken in the case of the visual indicator to ensure that it is not concealed by part of the system panelling or covering so that it is always clearly visible. Electrical indicators can also be installed at points which are difficult to access since the maintenance interval is indicated by an electrical signal which can be used for a number of purposes.

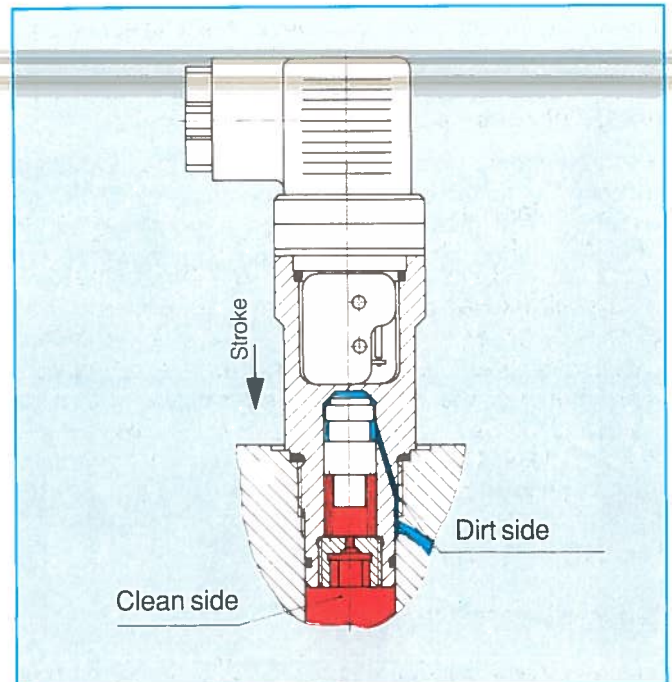


Fig. 22 Electrical differential pressure indicator

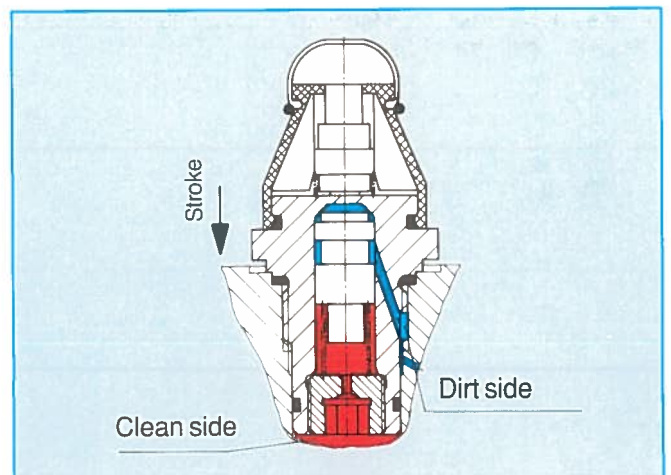


Fig. 23 Visual pressure difference indicator

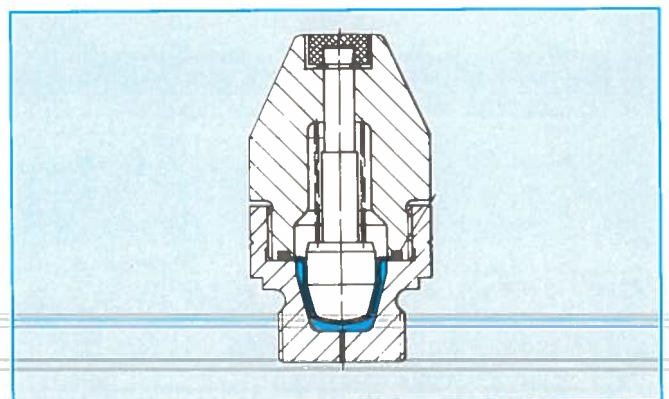


Fig. 24 Visual clogging indicator for return line filter

Electronic clogging indicators are available for special applications. Such indicators are successfully used particularly under dynamic operating conditions in conjunction with differential pressure-proof elements, as used at low starting temperatures or frequent pressure peaks.

This electronic indicator suppresses the indication function up to an operating temperature of 32 °C for example. The pressure peaks are suppressed for an effective period of 9 seconds and can therefore not trigger the indicator function.

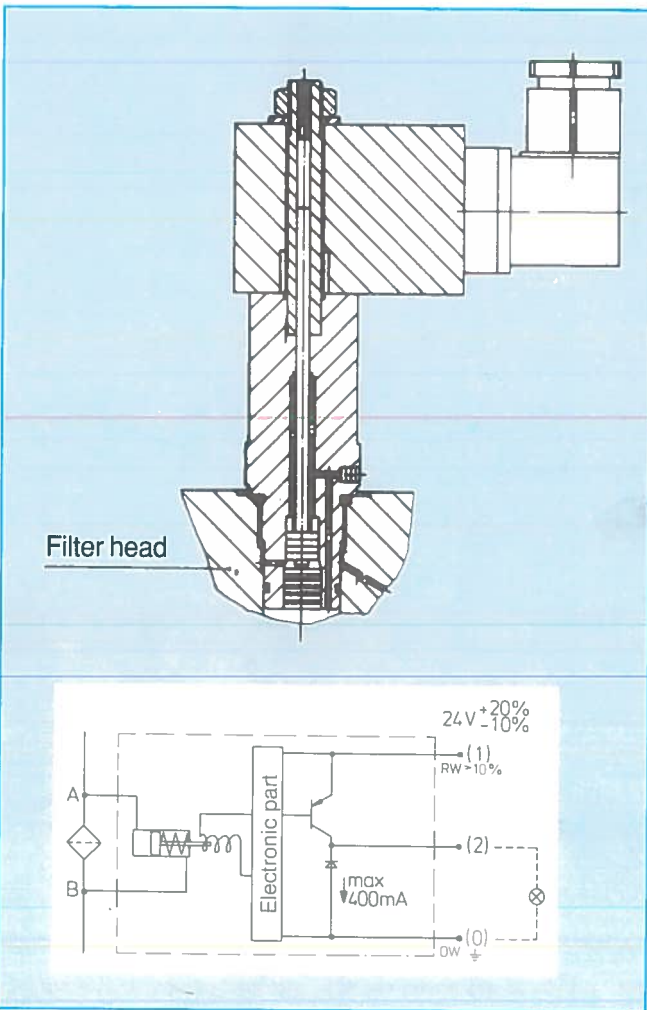


Fig. 25 Electronic differential pressure indicator

### Filter-Breather

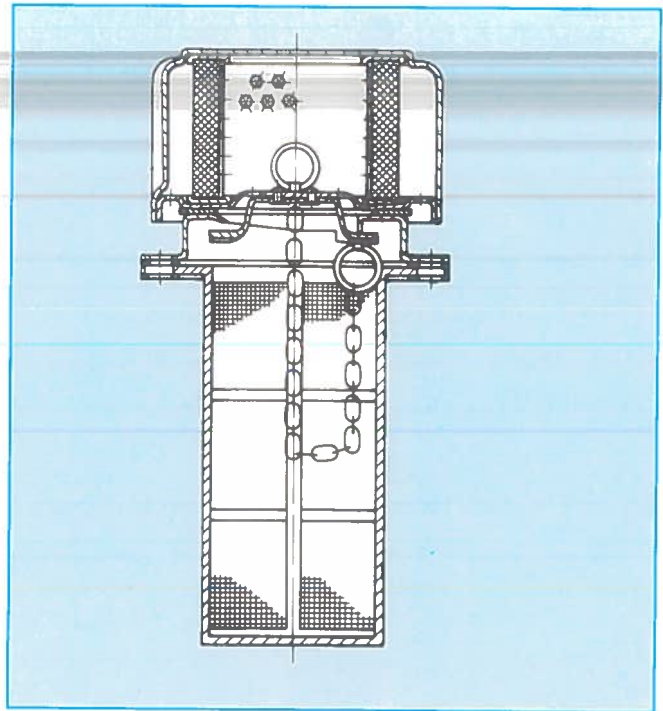


Fig. 26 Sectional view of a filter-breather

## Notes on Maintenance

### 1. Filling and Flushing the Hydraulic System

A further opportunity for dirt to enter the system from the outside is provided when filling the system with hydraulic oil. Due to its manufacture, filling, transport and storage, the new pressure medium can already at this stage be contaminated to a relatively high degree. In order to eliminate this possibility, it is advisable to fill the system via one of the filter units shown in *Fig. 2*. These types of filter units are also particularly suitable for flushing a system prior to commissioning. Flushing reduces the contamination incurred during installation to a degree necessary for reliable operation of the system without unnecessarily subjecting the filters of the system to additional load.

The size of the filler connection must be selected corresponding to the pump delivery.

The pore size of the filter element must be at least the same as that of the system filters.

To ensure quick handling of the devices shown in *Fig. 2*, it is recommended to provide quick-release couplings at the hydraulic tank.

### 2. During Commissioning

Check whether the fluid, pressure and flow of the system agree with the specifications for the filter provided in the brochure and on the filter.

### 3. During Operation

Open the filter housing and clean when indicator responds. Renew seal if leaks are found at the housing.

**Caution:** *Depressurize before opening filter.*

### 4. Changing Filter Element

- a) Generally, all filter elements should be changed after one year of operation.
- b) The filter element must be changed when the signal "filter clogged" is indicated
- c) When changing the filter element, no contaminated medium must enter the hydraulic system. Contaminated medium must be drained from the filter housing before changing the filter element.

Notes

Chapter L

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# **Practical Examples of Servo and Proportional Valve Systems**

Josef Hutter



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### Preface

The requirements which proportional technology must meet are becoming more and more demanding. The demands made on planners of hydraulic system with this technology are also increasing at the same rate.

In addition to thorough knowledge of device functions, several important criteria must be borne in mind when designing control circuits:

- The natural frequency of a system
- Correct spool size  
Pressure drop at the control lands
- The control range -  $Q_{\min}/Q_{\max}$
- The influence of changes in mass, velocity, pressure and viscosity  
Limits of time-dependent delay.
- Are pressure compensators necessary?  
Meter-in pressure compensator/  
meter-out pressure compensator
- Are deceleration or counterbalance valves necessary?
- Pressure intensification in single rod cylinders and meter-out pressure compensators  
Total pressure at motors
- Is an increase in the differential control pressure at pressure compensators appropriate or necessary?
- Is an open loop control at all possible or must a closed loop control be implemented?
- Selection of valves with sufficiently fast response for the relevant tasks, particularly in closed loop controls.

The following application examples from various sectors of industry represent a cross section of typical tasks performed by hydraulic systems. They clearly show that the above mentioned criteria have been implemented.

An essential factor regarding planning of open loop controls and drives in proportional hydraulics is precise definition of the intended task. With an exact definition of the application, it is possible - almost without exception - to define an optimum solution.

### Radio Control for a Suspended Mono-rail System in the Mining Industry

Cable-driven suspended mono-rail systems are used in the mining industry to transport material and personnel.

Hydrostatic drive systems have proved to be well suited to cable-driven transport systems due to their simple speed adjustment while maintaining the necessary tractive effort over the entire speed range.

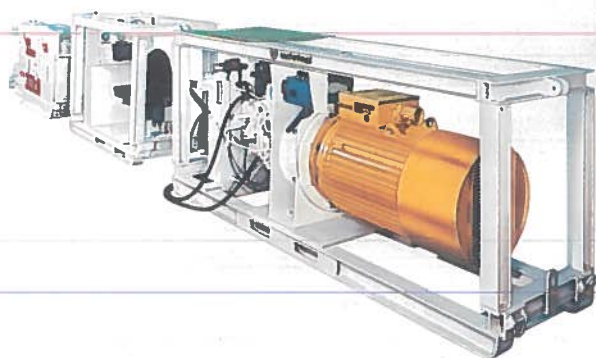
The flow delivered by the pump and therefore the speed of the system change dependent on the control pressure. The swivel angle of the axial piston pump is proportional to the pilot pressure of the pilot unit. To ensure the system is always operable, two systems are provided for the control of the pilot pressure to the pump:

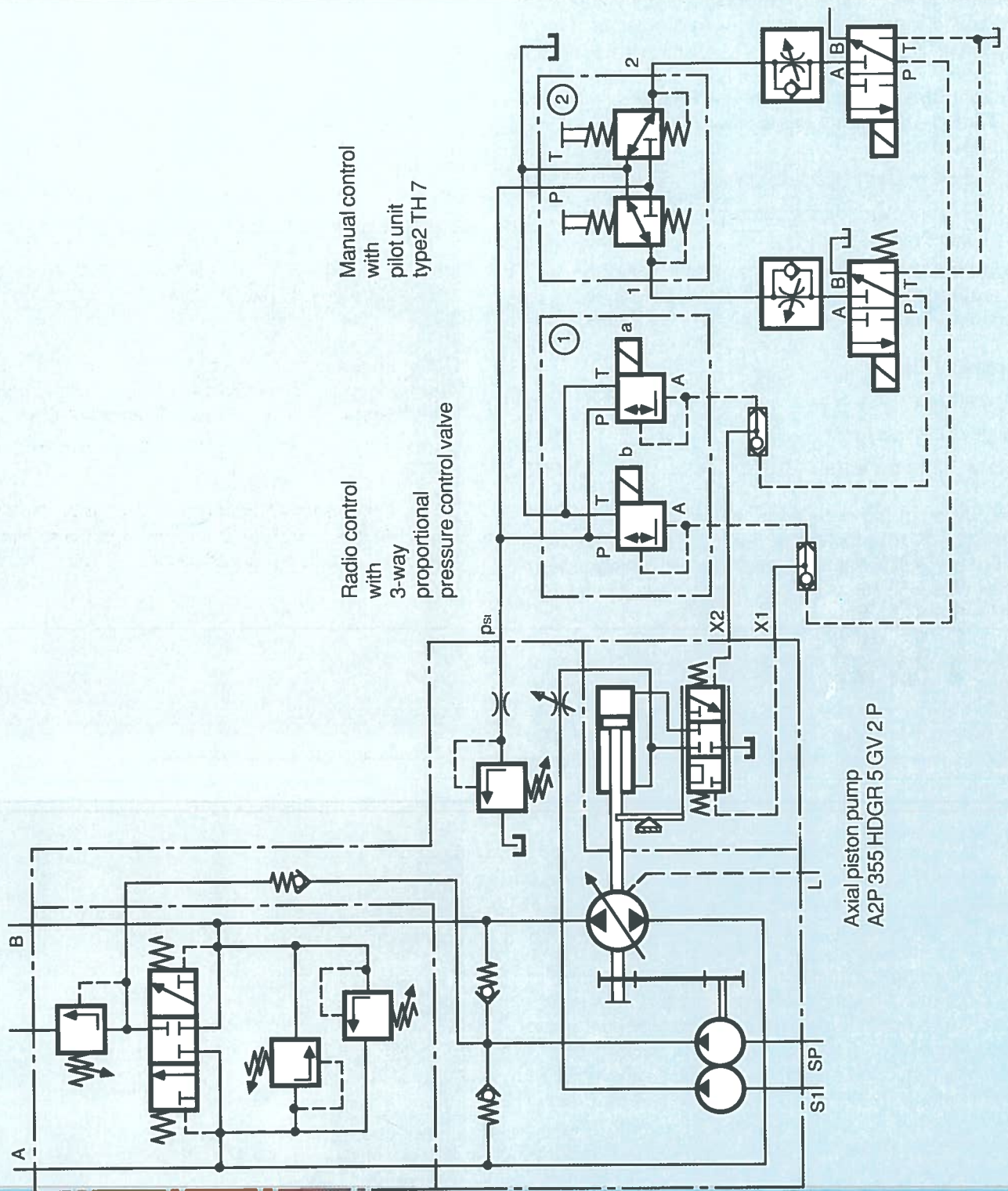
- 1) With a 3-way proportional pressure control valve 3 DREP 6 C (Item 1)
- 2) With a manually operated pilot unit 2 TH 7 (Item 2)

The pressure control valve is actuated by means of radio control. The driver operates a portable transmitter. The high frequency transmission between transmitter and receiver takes place in the 30 MHz frequency range. The frequency-modulated signals received are based on a digital modulation system, the so-called pulse code modulation (PCM) which, compared to other systems, offers the highest possible degree of transmission reliability.

During manual control, using the pilot unit 2 TH 7 from the driving position, the operator is linked with the driver of the train by radiotelephone. A coaxial cable running over the entire length of the system facilitates transmission of the radio signals.

Both the proportional pressure control valve 3 DREP 6 C and the pilot unit 2 TH 7 are modified units, and have been approved to BVS Specifications.





### Drive of Ladle Change Car in a Converter Steel Works

Converter linings are subject to wear and require regular replacement. For this purpose, the converter must be moved in various positions with the aid of a ladle change car.

#### A Ladle Change Takes Place in 4 Phases:

- 1) The ladle being changed is moved to its parking point
- 2) The car moves to the feed stand
- 3) The newly overhauled ladle is moved to the converter stand
- 4) The old ladle is moved from the parking point to the feed stand

#### Technical Data

Car diameter	16 m
Height of car with ladle	9 m
Total weight car + ladle	1200 t

The maximum travel speed of the car is 15 m/min. This corresponds to a speed of 3.2 rpm at the four drive wheels. The travel speed must be completely free of jolts with sensitive continuous control from almost zero to 15 m/min.

It must be possible to approach the individual positions with a relatively high degree of accuracy. At a crossing point, the car is turned through 90° about its centre. For this purpose, the complete car is raised and after being turned lowered onto the new pair of rails. During this operation, the positioning accuracy is  $\pm 30$  mm, quite accurate, considering the dimensions and weights involved.

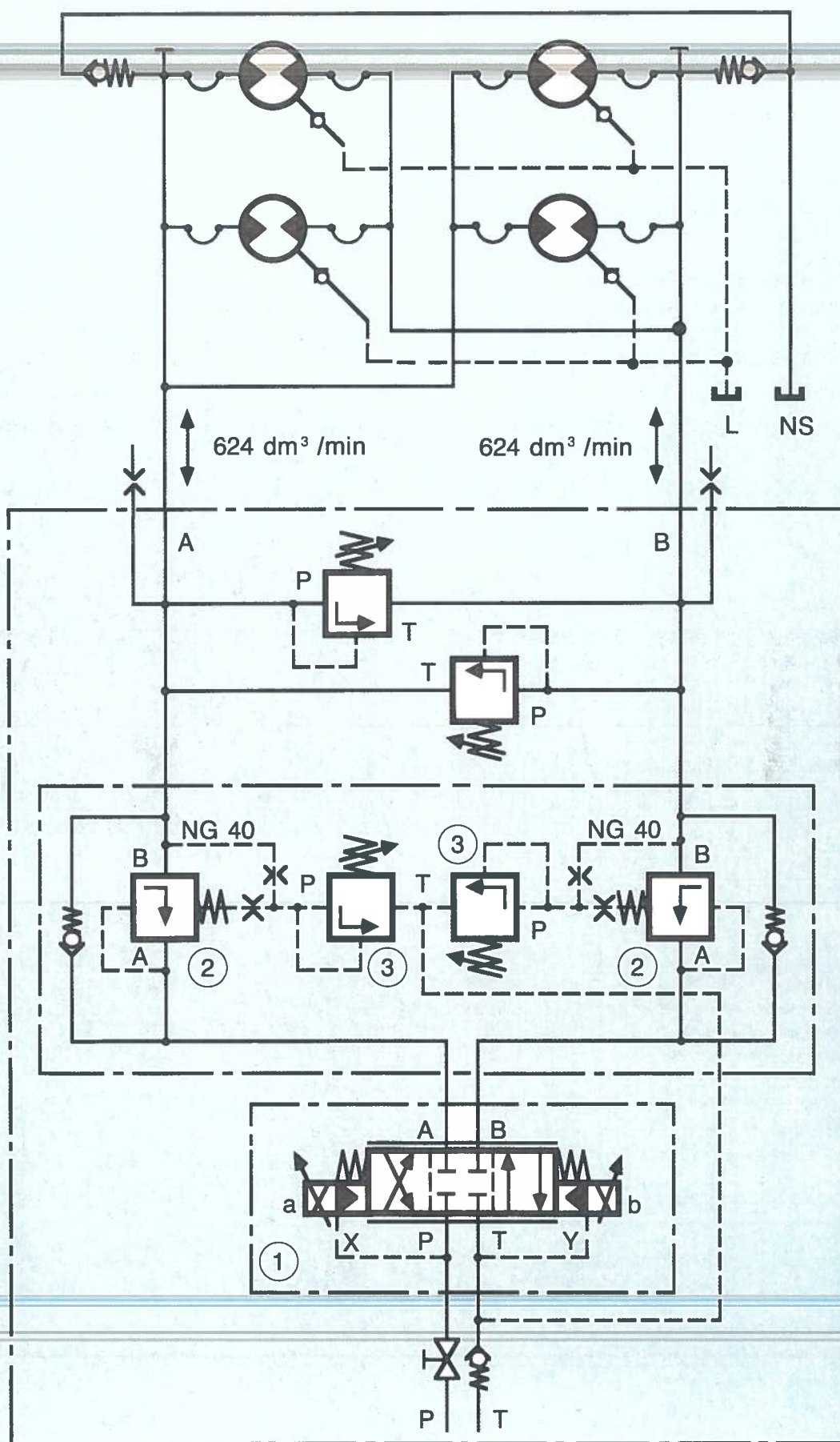
All drive procedures are controlled with the proportional valve (Item 1). Influences of varying rail friction, load viscosity etc. are compensated by means of meter-out pressure compensators in the lines A and B. These pressure compensators are integrated in the form of DR logics in the control manifold. Pressure relief valves (Item 3) installed in the cover enable adjustment of the  $\Delta p$  at the orifice (= proportional valve spool). This is necessary since the proportional valve, type 4 WRZ 32 cannot cope with the maximum flow of 624 dm<sup>3</sup>/min at a fixed  $\Delta p$  of 8 daN/cm<sup>2</sup> of a sandwich plate pressure compensator. A higher  $\Delta p$  at the orifice results in increased flow. The pilot control of the proportional valve is provided manually with a manual control unit. For the operator it is important that a certain angle of deflection of the control unit always produces the same speed. The meter-out pressure compensators constantly guarantee this, even when the above mentioned operating conditions change.



Hydraulic drive of ladle change car



The ladle change car is moved in position over the line crossing point.



## Elevating Drive in a Rolling Mill

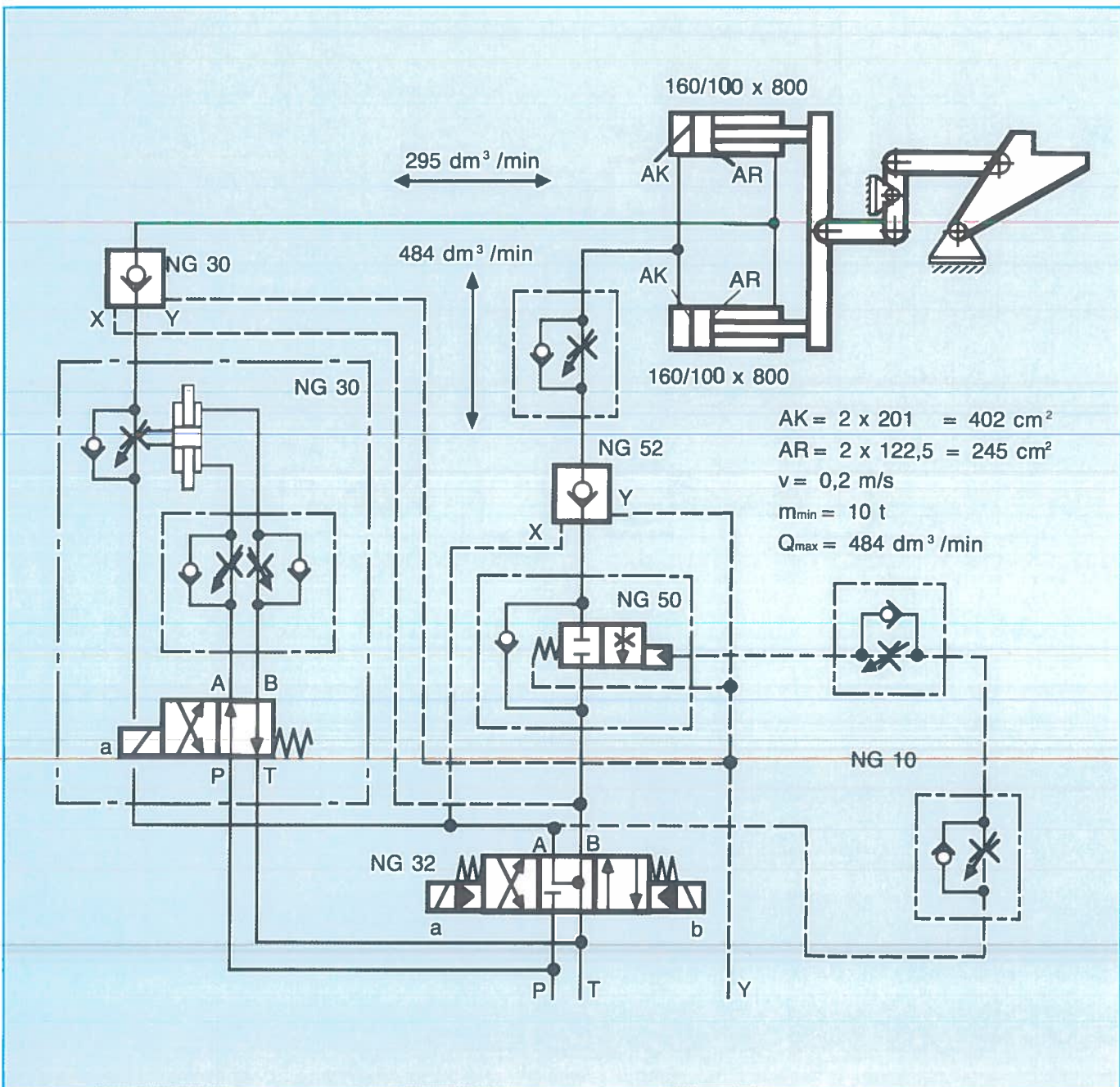
In the previous version, 8 devices were necessary for the control of the lift cylinder.

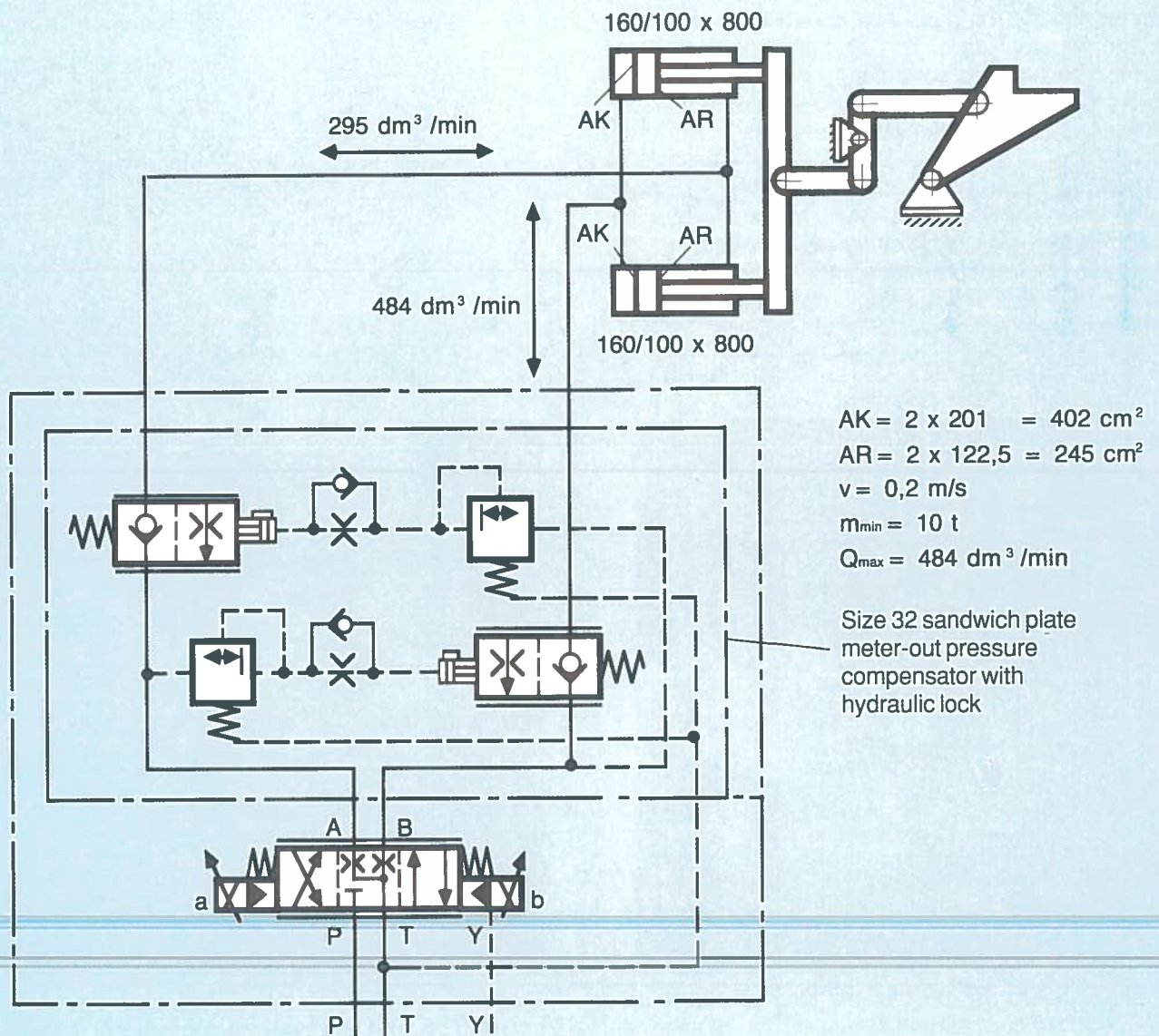
The individual piping of these devices, or mounting or installation in a control manifold was costly and complex.

Optimum adjustment and matching of all devices required considerable time.

In the new version with a proportional valve, only one device need be connected via the subplate, or mounted on a control manifold due to the fact that the meter-out pressure compensator is designed as a sandwich plate.

This meter-out pressure compensator contains two pressure compensators connected in the lines A and B which provide cut-out in the centre position of the proportional directional valve.





### Elevating System in a Welding Line

The welding line is used in the production of bodies for passenger vehicles. The installation has a total length of 30 m. All 12 lifting stations are lifted and lowered simultaneously by means of a corresponding elevating system. Material is transferred or deposited at the centre of the lift motion. The transfer speed must not exceed 0.15 m/s, otherwise the automatically positioned sheet metal parts would be ejected. On the other hand, the lifting and lowering cycle should be completed as fast as possible.

These requirements are met by a proportional valve together with electronic devices for position-dependent deceleration.

Electronic proximity switches, so-called analogue initiators are moved along metal cams. The output voltage is reduced to 0 V as the proximity device approaches the cam. This voltage is fed to a amplifier specially designed for this purpose and controls the solenoids of the proportional valve. This arrangement does not represent a closed loop control but rather a position-dependent open loop control, which is position-dependent during the deceleration phase.

As shown in the example, the speed can be reduced to any value and increased once again to the initial value in any arbitrary position by means of a cam. The decisive factor in this arrangement is the distance X between the cam and the connecting line of the two end cams.

Since analogue determination of position is only effective in the area of the deceleration path, installations of any length, e.g. drive systems can be equipped with this equipment.

This technology is primarily used when a drive must approach a position with a relatively high degree of accuracy under conditions of varying kinetic energy.

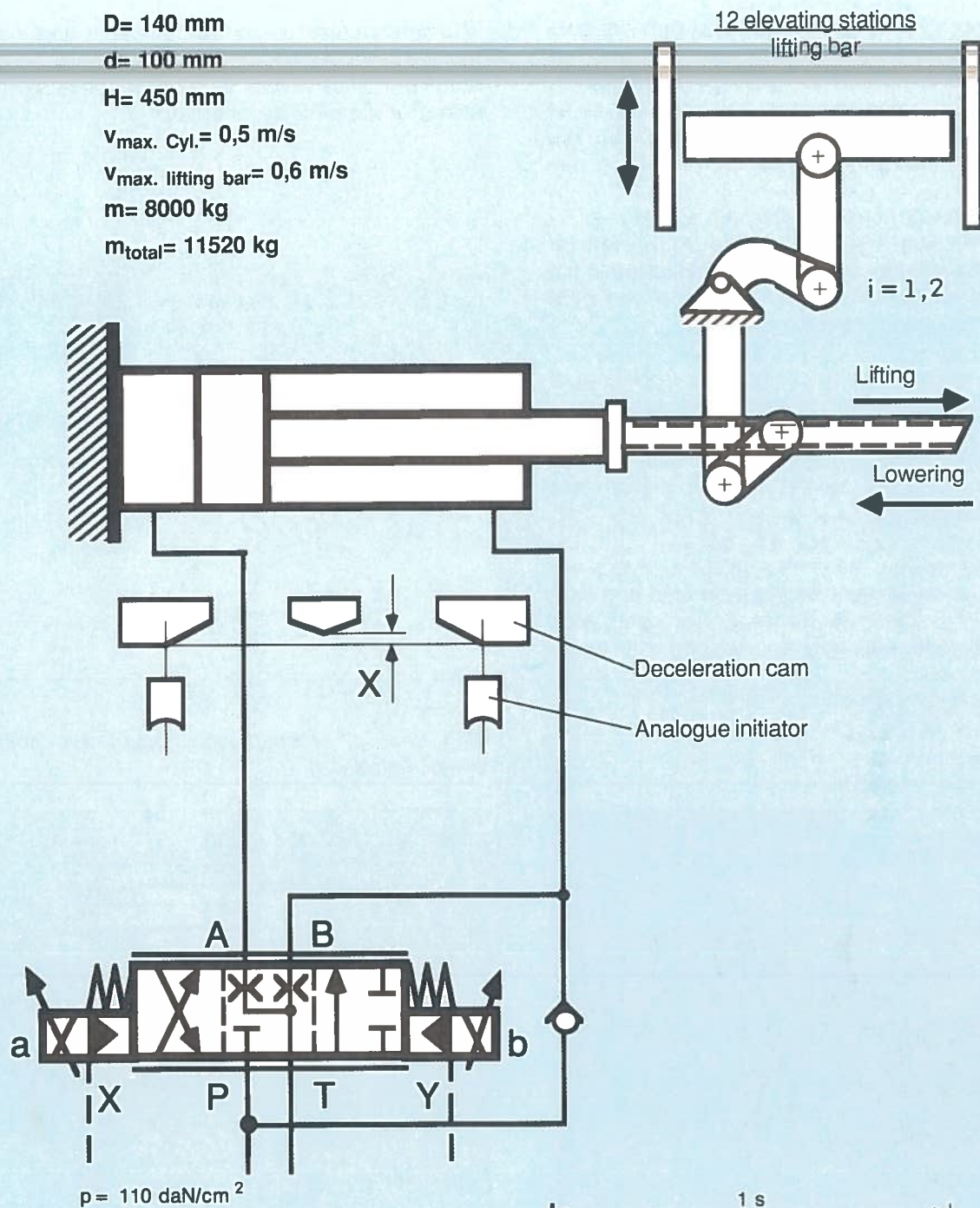
Time-dependent deceleration should be preferred if the speed of a drive is higher than approx. 1 m/s.



*The accumulator unit on the left-hand side provides the 460 l/min of hydraulic oil necessary for the acceleration procedure. The Type V4 vane pump to the right fills the accumulator during the "no-movement phase". The proportional directional valve Type 4 WRZ 25 is*

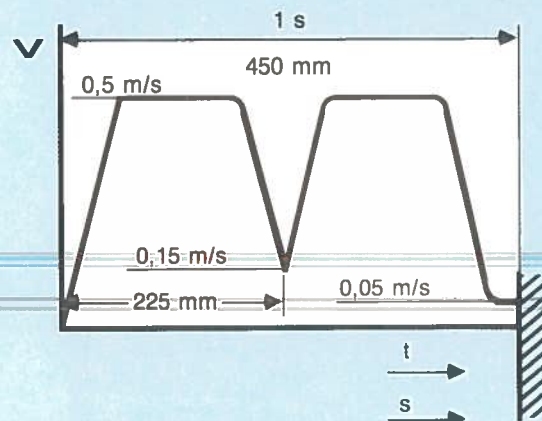


*One cylinder - the other serves as a standby - moves in conjunction with a lifting mechanism, all 12 stations simultaneously.*



4 WRZ 25 W3-325-30

Block diagram of a hydromechanical drive  
 for a welding line  
 and its motion sequence diagram



## Chain Conveyor - Drive Cylinder

In a hot-rolling mill, the coils at the end of a hot conveyor line must be conveyed to a storage point. The temperature of the coils at the take-up reel is approx. 800 - 1000 °C. During transport, the coils are to be cooled to a temperature of approx. 500 - 600 °C. The conveyor chain frequently runs outside the bay in outdoor sections.

The total length of the coil conveyor chain is 280 m.

At a transfer station, the coils coming from a shorter chain driven by a hydraulic motor are transferred to the chain running at floor level. This chain is driven by a drive cylinder at a cycling stroke of 3600 mm.

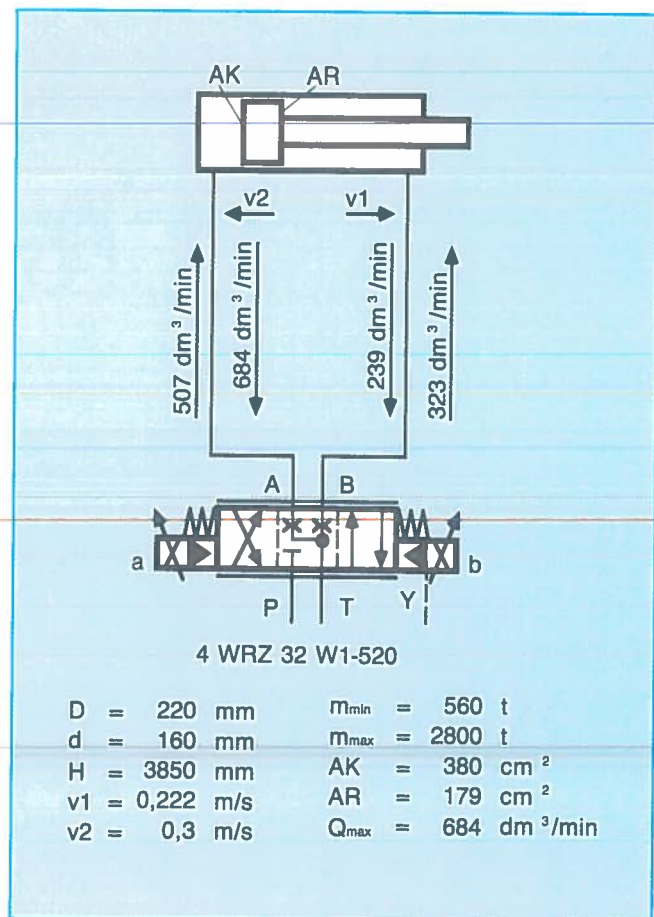
The drive is engaged in the chain at the beginning of the stroke. On completion of the stroke and disengagement of the drive, the chain stops during the return stroke. After return to the initial position, a new cycle begins as soon as the transfer station has transferred a new coil.

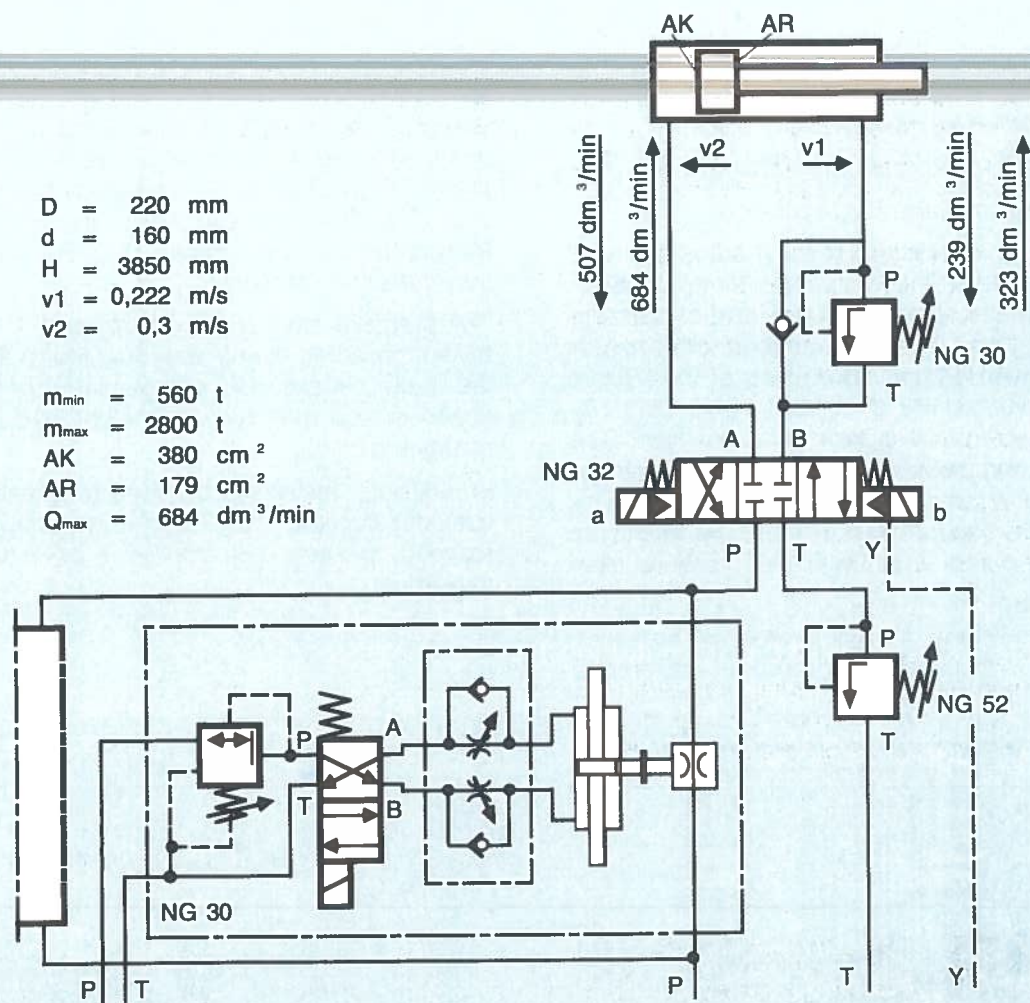
In the previous version, the control involved several devices. Optimum adjustment was complicated and time consuming. With the new concept, the open loop control is achieved with only one proportional directional valve.

This arrangement is considerably less expensive and much easier to handle. The acceleration and deceleration ramps as well as the speeds are easily adjusted at the front panel of the amplifier.



*New concept of open loop control with proportional directional valve*





Previous version of open loop control

## Open Loop Control for Airfreight Elevating Platform

The open control must comply with the following conditions:

- jolt-free acceleration and deceleration
- load-independent speed control in all drive phases
- low losses during constant pump operation.

Constant speed, irrespective of the load, is achieved during upward movement with the 3-way meter-in pressure compensator (Item 3) in the form of pressure limiting logics. This pressure compensator has a control spring of 4 daN/cm<sup>2</sup>. Unloading of the system in neutral is achieved by directional valve (item 6), allowing the pump to flow to tank at 4 daN/cm<sup>2</sup>. The pressure relief valve (Item 5) in the load pressure line permits variation of the differential control pressure. The setting in this case is 10 daN/cm<sup>2</sup>. The maximum pressure of the pump is set with the pressure relief valve (Item 4).

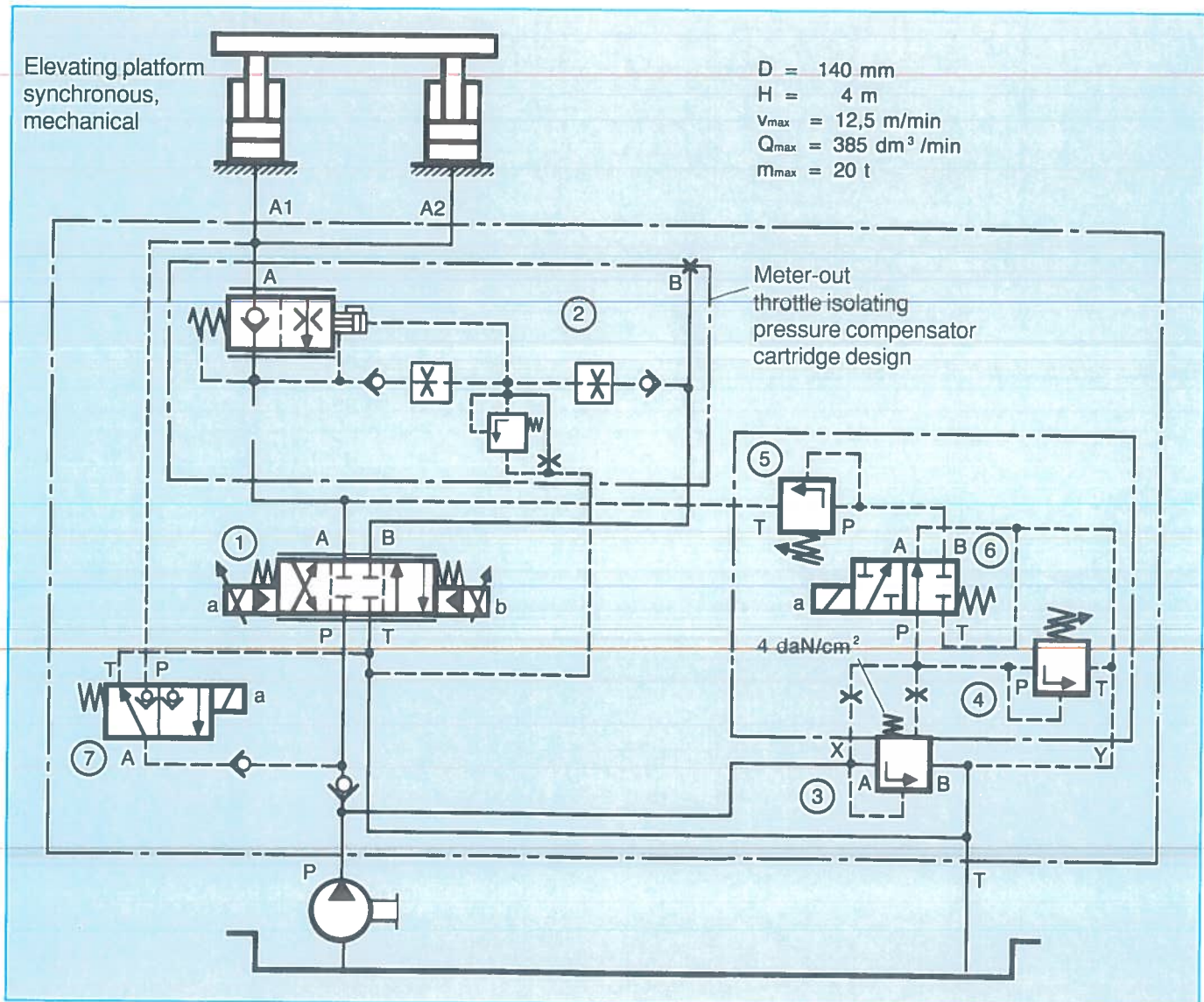
During upward movement, the pump pressure is automatically set to the required load pressure + differential

control pressure of 10 daN/cm<sup>2</sup> at the orifice – control land P to A.

When the lift is stopped and during downward movement, the electric motor of the constant displacement pump is switched off. The directional poppet valve (Fig. 7) is therefore required for the pilot oil supply of the proportional valve (Item 1) of the meter-out pressure compensator (Item 2).

The pressure drop at the control land A to T is maintained constant during the downward movement by the meter-out pressure compensator. In this way, the speed is also maintained constant irrespective of the changes in load.

In addition, the meter-out throttle isolating pressure compensator also provides leak-free cut-off with the lift stopped, and acts as a non-return valve during upward movement.

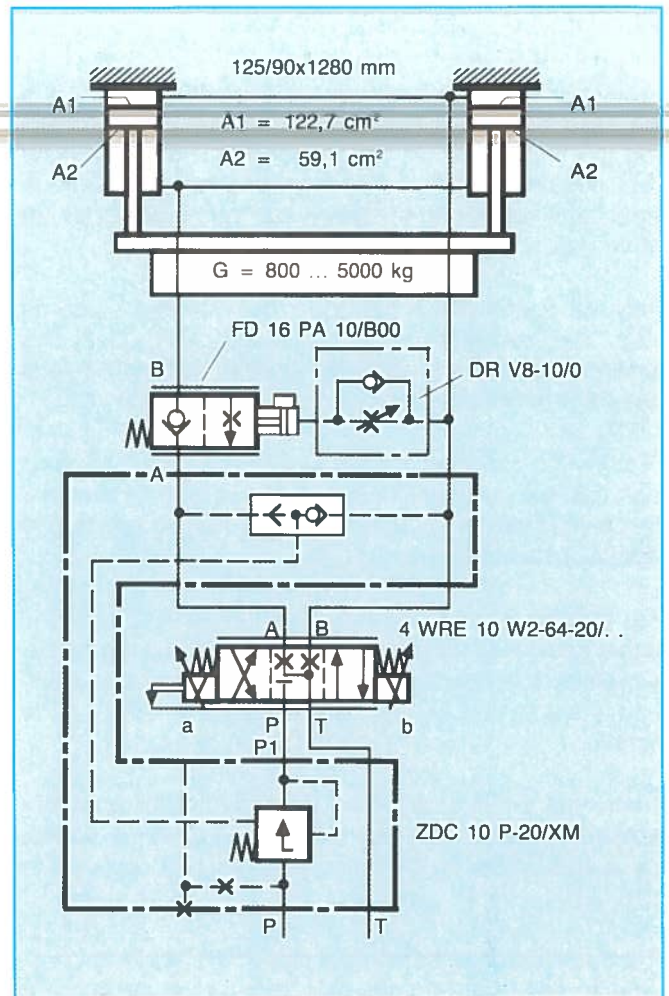


## Stacking Device for the Paper Industry

A two-way meter-in pressure compensator (sandwich plate) is used in order to avoid pressure intensification at the single rod cylinders.

The negative (pulling) load makes the installation of a counterbalance valve necessary. During downward movement, the load must be taken up by the counterbalance valve to ensure the pressure drop from P to B at the proportional directional valve remains constant at 8 bar.

The load pressure is sensed from the load lines with the aid of a shuttle valve. This shuttle valve is integrated in the pressure compensator.



## Manipulater Car for 2 Presses

Gas bottles are produced on a pair of hot forming and hot drawing presses. Transport between the presses and the feed runs fully automatically. The manipulator car - consisting of the upper car for longitudinal movement and lower car for entry and exit - performs all movements.

Only the operations of the upper car will be considered here. The maximum travel distance is 6 m. Over this distance, 5 positions must be approached with a relatively high degree of accuracy.

The drive is provided directly by a hydraulic motor, via a rack and pinion. A pilot operated proportional directional valve (Item 1), Type 4 WRZ 16 E 100 is used for the control of the movements.

The electrical control of the drive is achieved with the digital positional amplifier VT 4630. With this amplifier, hydraulic drives with proportional or servo valve open loop controls are positioned digitally by BCD code signals.

Before the required position is reached, a distance dependent deceleration function is engaged, in which, the output signal to the valve is gradually reduced to zero.

The amplifier enables internal position presetting via 5-position decade switches, or externally by means of a freely programmable control. The 5-segment position display can also be provided internally or externally.

When setting the positions via a PC system, the number of positions is optional. When set internally, the number of positions is limited to 9.

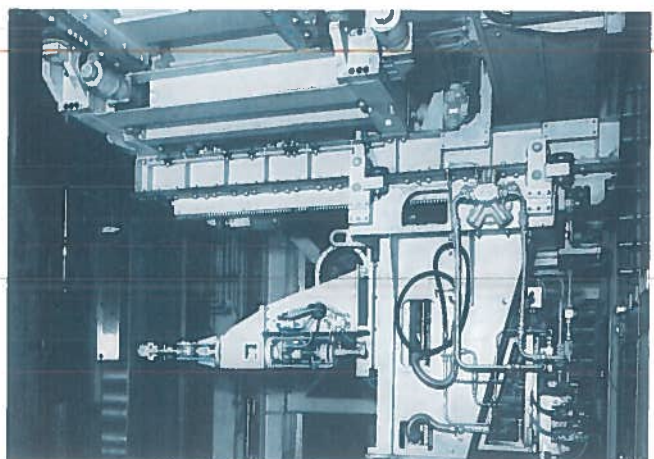
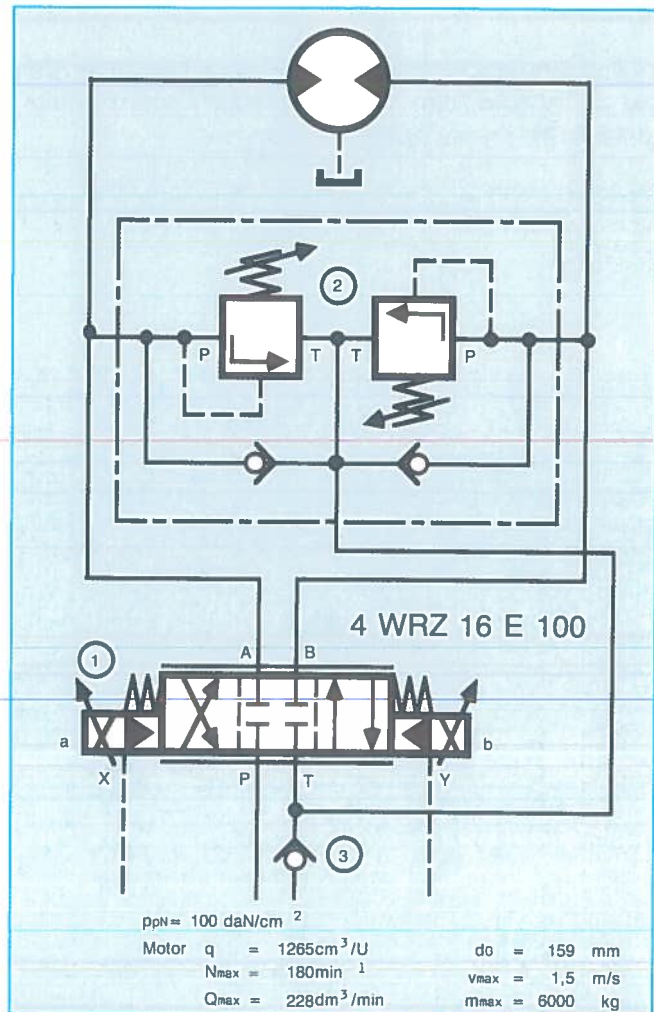
The speeds, acceleration ramps, deceleration ramps (for inching) and deceleration distance for distance-dependent deceleration are set by means of the potentiometers on the front panel of the amplifier. Pre-setting the 5 positions to be approached takes place externally with the PC control.

In this case, the position is determined with an incremental shaft encoder at 1250 pulses/revolution. At a pinion diameter  $d_0 = 159$  mm, 2 revolutions = 1 m travel distance = 2500 pulses. This pulse count is quadrupled in the amplifier. In this way, 1 m travel distance is triggered in 10,000 pulses (1 pulse = 0.1 mm) thereby providing the required positioning accuracy of  $\pm 1$  mm.

10,000 pulses = 0 ... + 10 V are made available on the amplifier for deceleration. The calculated deceleration is 0.75 m = 7500 pulses.

During an "emergency stop", the proportional valve is no longer controlled via timed ramp or distance dependent deceleration, but is allowed to close in its own natural minimum closing time (approx. 70 ms). In order to protect it under these conditions, the relief and anticavitation valve (item 2) has been installed.

To ensure filling of the feed side, it is good practice to install a non-return valve (Item 3) with a 3 bar cracking pressure in the tank line.

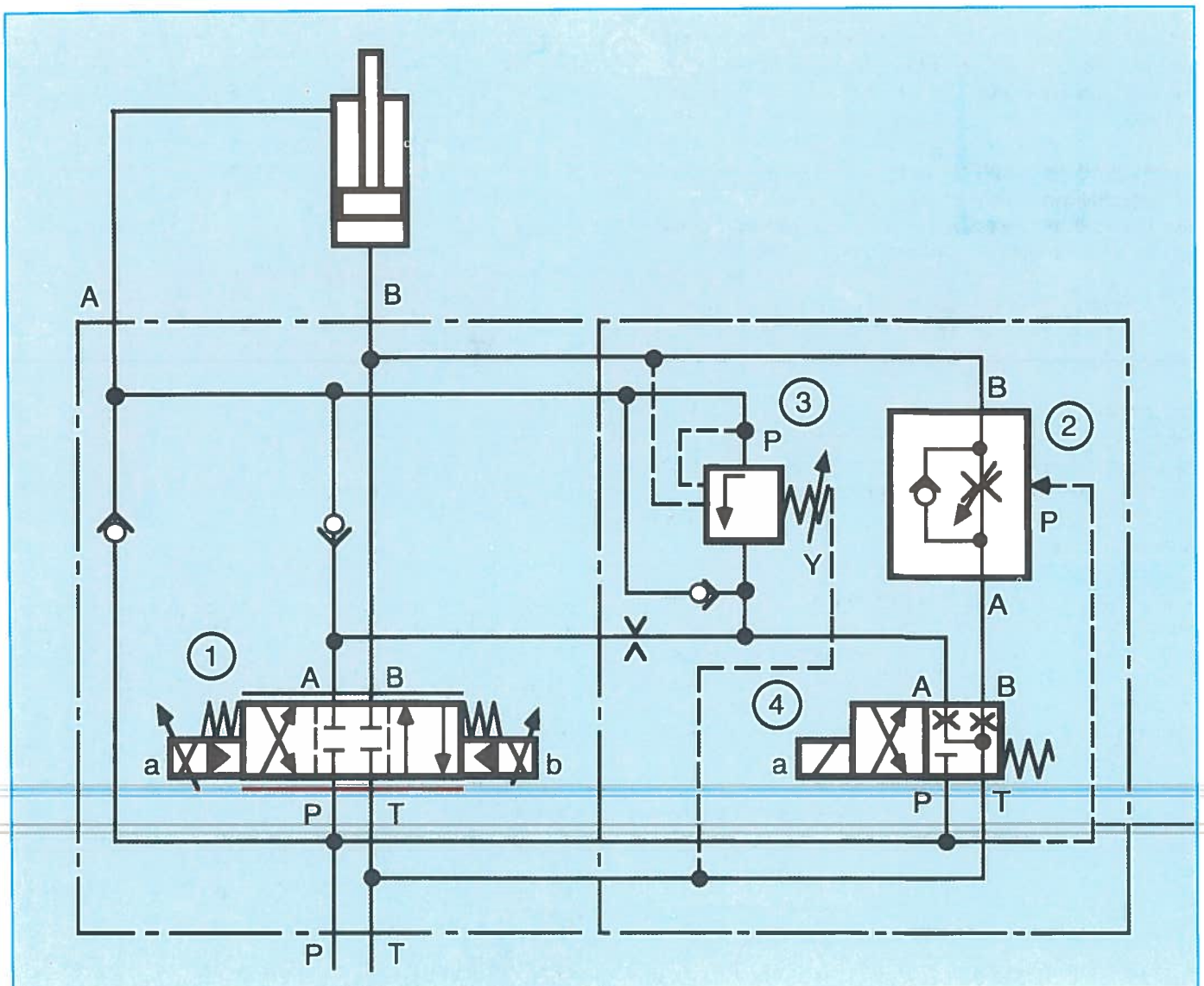
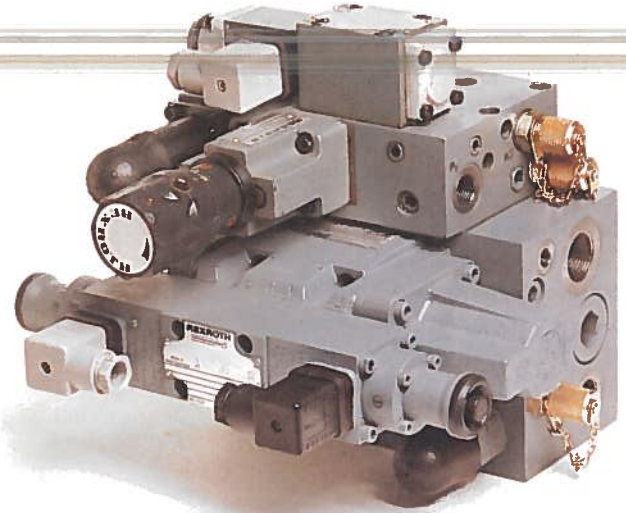


## Slide Unit

Slide units in transfer lines mainly feature cylinders with an area ratio of 1:2. A regenerative circuit is used in these cases. Compact control manifolds in the sizes 6, 10 and 16 of modular design are directly mounted on the unit cylinders. The proportional directional valve (Item 1) as a rapid traverse valve enables shock-free starting and deceleration of relatively large kinetic masses. Rapid traverse speeds of up to 25 m/min are often realized in time-dependent units in transfer lines. With the current regulator (Item 2) the feed rate is set by conventional means.

An optimum backpressure is set automatically at the load-dependent retaining force valve (Item 3) in each phase of the operating cycle.

The rapid traverse rate as well as the acceleration and deceleration values can be easily set at the electrical amplifier.



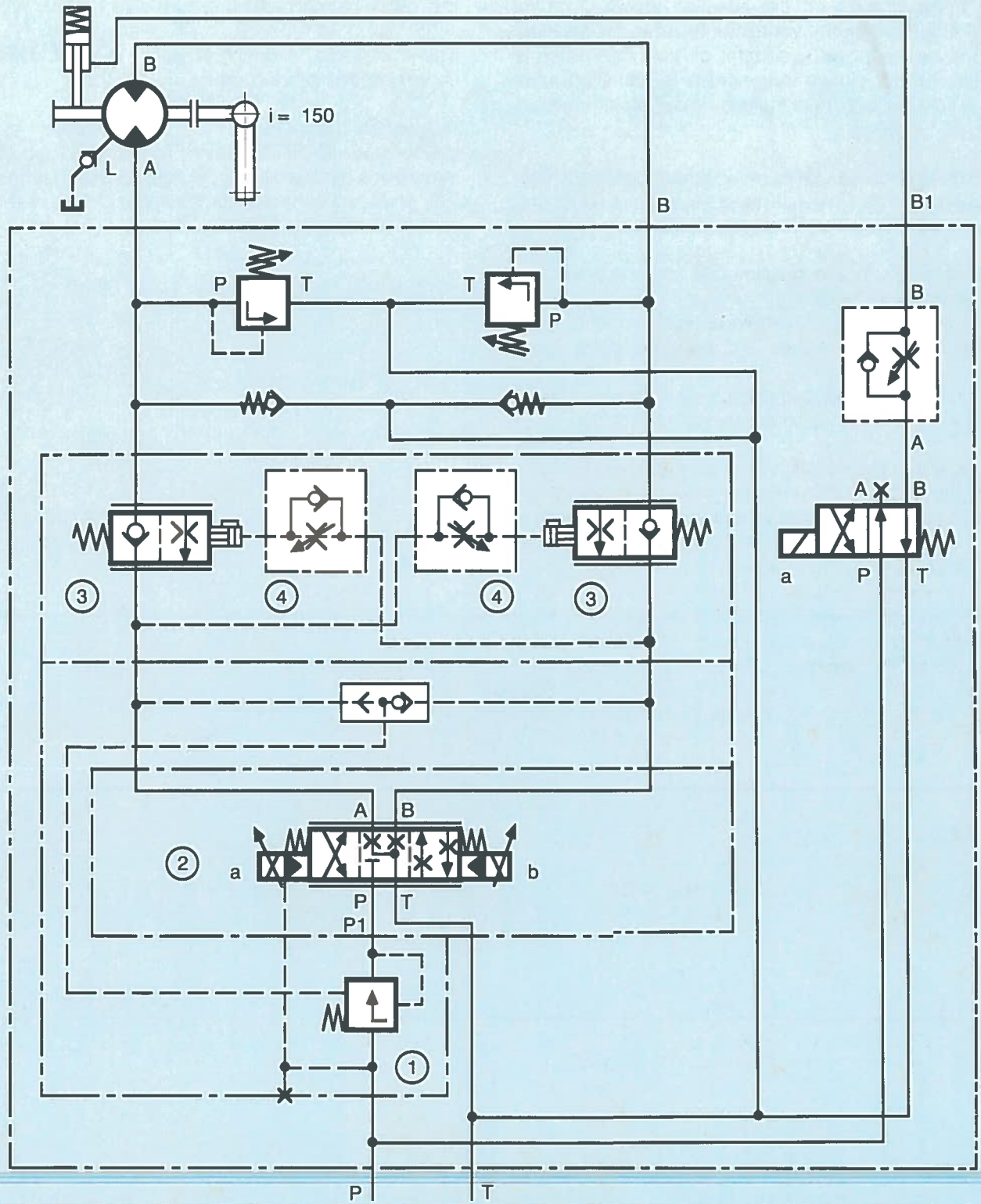
### Rotary Drive of Platform

The rotation (inclination) of the platform about its centre axis is provided by a hydraulic motor in conjunction with a worm gear with a ratio of 150:1. The rotary speed of the platform must be continuously variable from almost zero to 1 rpm. For this reason, an axial piston low-speed motor, Type MCS is used. Under given conditions, this motor ensures low torque fluctuations and low pressure fluctuations - a minimum smooth speed of 0.5 rpm.

A meter-in pressure compensator (Item 1) is connected upstream of the proportional directional valve (Item 2) to provide the necessary load compensation. It is not possible to use a meter-out pressure compensator since the motor would be subject to excessively high load at the operating pressure of 150 daN/cm<sup>2</sup>. The permissible summated pressure of the motor is max. 300 daN/cm<sup>2</sup>. The acceleration pressure which occurs during the acceleration phase must be added to double the operating pressure. The maximum permissible summated pressure would then be exceeded.

The meter-in pressure compensator provides load compensation = holding  $\Delta p$  constant at the orifice, only when the load direction is positive. For this reason, deceleration valves (Item 3) are installed in the drive lines A and B. A further task of this device is to provide, for safety reasons, leak-free cut-off when the drive is stationary. To ensure the platform is reliably held in any position, inspite of internal leakage at motor, the motor is equipped with a hydraulically operated disc brake.

The control of the drive, i.e. actuation of the proportional directional valve is via a manual control unit.



## Injection Moulding Machine

The high demands placed upon modern injection moulding machines regarding the quality of the finished parts, makes it ever more necessary to employ closed loop control on the injection process. Using closed loop control, the variations found in the finished products are reduced by a factor of nine. Optimising a machine having closed loop control is achieved after only a few cycles, thus quickly ensuring production quality.

A further increase in quality, even in complex parts can be achieved when internal die pressure measurement is integrated in the closed loop control circuit.

The progression of the injection rate curve is determined based on process data.

The stroke of the injection cylinder is determined by a position measuring system and correspondingly processed.

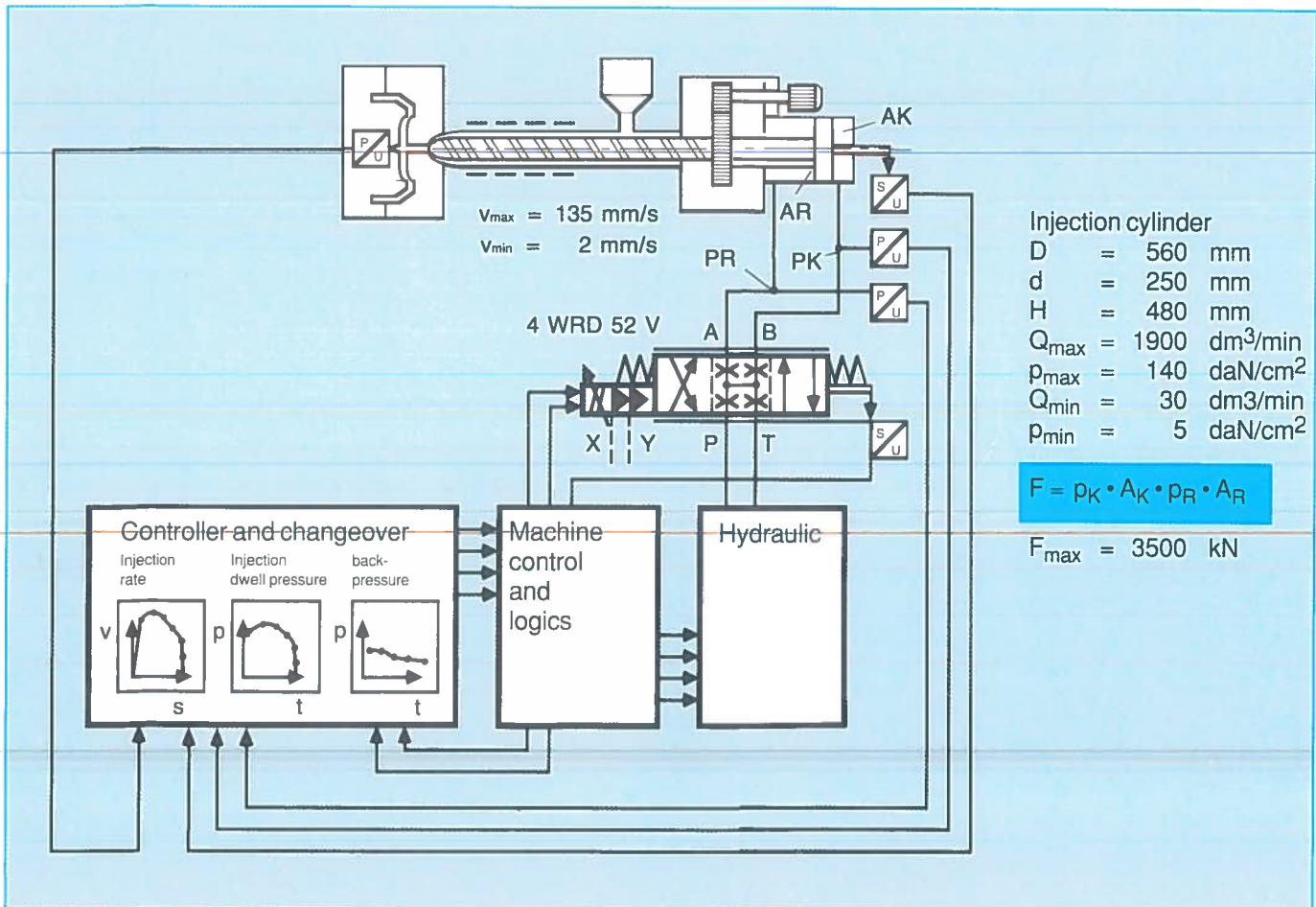
The actual value produced in this way is compared with the signal value of the injection curve and corrected.

With regard to the injection or dwell pressure and providing the internal die pressure measuring system is used, the internal die pressure can precisely follow a preset dwell pressure curve independent of the viscosity of the melt.

The changeover from closed loop speed control to closed loop pressure control can take place either dependent on the "injection stroke" or dependent on the "mass/internal pressure".

The backpressure during plastification also follows a curve based on process data.

All control procedures are implemented by the control valve, 4 WRDE 52 V. The closed loop control electronics is based on microprocessor technology. The analogue controller for the valve is designed in the form of a hardware controller.



**Notes**

Notes

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## Key Word List

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Meter in pressure compensator, 2 way C2	Proportional pressure reducer valve
Meter in pressure compensator, 3 way C6	(pilot operated) B22,B24
Meter out pressure compensator C10	Proportional pressure relief valve
Meter out pressure compensator, cut off function C11	(closed loop) B13
Multi-pass test K4	Proportional pressure relief valve
Natural frequency E10	(direct operated) B18
Natural frequency, double rod cylinder E23,E24	Proportional pressure relief valve
Natural frequency, hydraulic motor E26	(pilot operated) B20,B21
Natural frequency, influence of E22	Proportional pressure valve A1,A5,B18
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